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LOCOMOTIVE ENGINEERING,

AND

THE MECHANISM OF RAILWAYS:

A TREATISE ON THE PRINCIPLES AND CONSTRUCTION

OF

THE LOCOMOTIVE ENGINE,

RAILWAY CARRIAGES, AND RAILWAY PLANT,

WITH EXAMPLES.

Illustrated by Sixty-four Large Engravings and Two Hundred and Forty Woodcuts.

BY ZERAH COLBURN,
CIVIL ENGINEER.

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PREFACE.

The circumstances under which this work has been prepared appear to require some explanation. Mr. Colburn, an Engineer of great attainments, great abilities, and remarkable force of character, commenced the work, with the additional advantages of an extensive connection in the railway world, both at home and abroad; and there is no doubt that, had he lived to complete it, he would have done ample justice to the comprehensive programme laid down in his Introduction. He was removed, however, in the midst of his career; and, owing to the pressure of other engagements, he had already engaged the assistance, under his immediate direction, of Mr. W. H. Maw, a thoroughly qualified Locomotive Engineer; Mr. Slade, an American Engineer; and Mr. Ferdinand Kohn, an Engineer who has made his mark in the history and progress of iron and steel manufacture. These gentlemen contributed the chapters to which their names respectively are attached in the body of the volume. On the death of Mr. Colburn, however, the writer was requested by the publishers to conclude the work, which had been completed up to the end of Chapter XV. For the remaining chapters, at the commencement of which the writer's name is attached, he is responsible, and also for the general matter of Indices and Contents.

The Plates and Woodcuts which accompany the letterpress have all of them been executed with the completeness and accuracy that the subjects demanded; and, thanks to the courtesy of the numerous Engineers who supplied the original drawings, the Locomotive Practice of the present day, not only in England, but also on the Continent, and in the United States, is amply represented. In all cases the characteristics of the working drawings from which the reductions were made, have been maintained as far as possible, in order to preserve, to some extent, the specialities in style of the originals.

It is believed that the work is such as would have received the approval of Mr. Colburn, had he lived to see it completed; and it is confidently hoped that it will satisfy the just expectations of its numerous subscribers.

D. K. CLARK.

11 Adam Street, Adelphi,
London, *March, 1871.*

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INTRODUCTION.

At least five classes of readers are likely to turn to a Treatise upon the Locomotive Engine. There are those who desire general information concerning the Locomotive, in common with other practical subjects; there are those who wish to know how to manage, and others who seek to qualify themselves to construct locomotives; there are a goodly number who look upon railway stock with reference to commercial considerations only; and there is a distinct class of persons, of a philosophical or inventive turn, who prefer to see the subject dealt with as a branch of strictly scientific inquiry, and with reference to ultimate improvement, or, if it be possible, perfection. Besides these classes of readers there is here and there one whose ambition is to know all that can be known of the whole subject.

That kind of knowledge of the Locomotive Engine which answers the purpose of a well-informed man has already become so popular that it almost amounts to ignorance to be without it. Locomotive mechanism is very simple in its elementary nature, and the mind is naturally disposed to receive and retain any adequate explanation of striking phenomena, whether mechanical or otherwise; and hence it is that there are now thousands of persons who, although in no way concerned in the construction or working of railway engines, are nevertheless competent to give a fair general explanation of their structure and mode of action. But popular knowledge is of necessity inexact; and although it has its uses—and there is no desire to exclude or to undervalue it here—almost all that could be said of the Locomotive in a popular lecture would be inapplicable to any practical purpose. As to a knowledge of the management of locomotives, the confession must be made, however unwillingly, that engine-drivers, as a class, owe nothing of what they know to books. They are neither required nor expected to read more than their time bills and printed regulations; and, indeed, a distinguished engineer once declared, apparently with considerable satisfaction, that his best engine-drivers could not read at all. Without discussing such a circumstance, it is to be remembered that there is a large class of amateurs always studying engine-driving as if it were a profession, and it would not be right to disappoint them in a book like that now before the reader. A knowledge of the whole routine of engine-driving is indispensable, too, to a numerous class of railway officials; and although there may be but very few actually in commission who would willingly own that they had derived any part of their knowledge of such a matter from books, yet it is certain enough that there are many others, now seeking railway appointments, who will not despise even the resources now offered them for the purpose in question. As to the construction of locomotives, the very best book which could possibly be written could only serve as an adjunct to that practical experience and general personal acquaintance with the subject which alone justify an attempt at responsible practice. A constructor will refer to any work which faithfully records what has been and is being done by others in his branch of construction; but he will, if he be qualified for his business, resist any pretensions to authority on the part of the author of such work. In the trial, not long since, of a cause growing out of a locomotive boiler explosion, the locomotive superintendent of one of the principal lines of railway in the kingdom was confronted with a quarto volume popularly supposed to be an "authority" upon the Steam Engine. "You are aware, sir, what this author says upon the strength of boilers, and of the proper mode of testing them, are you not?" said the plaintiff's counsel. "No," was the reply, "I have never read the book; I only know of its existence." Such an answer should not be lost upon an author ambitious of stamping his own views and preferences upon the minds of others. It is not wise to attempt too much, however self-assured he may be; and there is always the satisfaction of knowing that whoever will faithfully describe the actual practice in any profession, especially in that of Mechanical Engineering, may count upon a respectful hearing on the part of even the highest masters. The present work may reasonably be trusted to find favour, therefore, in this respect; for nearly every locomotive maker of eminence in the kingdom, and the locomotive superintendents of the principal railway companies, have already kindly placed at the Author's disposal everything in their practice which can be of service to him in his work. It can hardly amount to presumption on his part to believe that he will succeed in making good use of his opportunities, for he has, as a responsible manager and engineer, himself gone through the whole details of locomotive construction and working.

Commercial results are now the chief object of Engineering, and the Author has no sympathy with that affectation which would exclude the consideration of such results, and a special reference to mechanical economy, from even a professional treatise. The tenderness with which most engineering authors touch upon commercial considerations arises, possibly, from an over-fine estimate of the abstract scientific character of Engineering, and from an analogy, needlessly magnified, between its profession and that of medicine. To an amateur the comparison of a steam engine to a sentient organism is perhaps as pleasing, and indeed as natural, as it is unnecessary. One author, indeed, treats the Locomotive Engine "anatomically" and "physiologically;" another insists upon giving it "lungs," "arteries," "muscles," and "limbs," and so on to the end of a long list of analogies as readily conceived as they are foreign to any useful consideration of the subject. There may be no great risk incurred in stating here, as a fact, that the imaginative faculty is less and less exercised exactly as a practical acquaintance with an engineering subject is extended. And any indulgence of this faculty is attended very generally with the disadvantageous cultivation of prejudices to which the amateur, alive to the "poetry" or æsthetic claims of the subject, is often ready to sacrifice useful proportions or arrangements. These conceits as to the personal attributes of locomotives may be encouraged to any extent, and with the most absurd results. One individual will insist upon an analogy between a locomotive and a race-horse, or a stag, and will not be satisfied unless there is a visible suggestion of one animal or the other; another amateur recognizes the elephant, the ox, or the hippopotamus as the true type, and seeks to impress some semblance of one or the other of these beasts upon a 30-ton goods engine, and so on with other fancies. Even the ugly locomotive chimney employed in America for wood-burning locomotives is admired by some for the "air of frontal grandeur" and the "majestic expression" which it imparts to the machine beneath and behind it; and this single fact will show that anything may be admired—*de gustibus non est disputandum*—even to the tattooing upon a New Zealander's cheeks. If the Author may presume to record any personal advice in these pages, it is that none who aspire to become engineers should encourage any play of imagination involving the forms or proportions of mere mechanism, but that they should apply themselves solely to the consideration of the best mechanical means by which any mechanical purpose under consideration may be accomplished. Locomotive Economy, in its broad commercial interpretation, should ever be the sole object of Locomotive Engineering.

Turning now to the scientific principles embodied in the Locomotive Engine, and which to many minds form the sole grounds of its claim to the dignity of consideration, it does appear that it has been subjected to over-theorizing, and that unwarranted exactness has been assigned to its proportions. It is a mistake to suppose that any one scale of definite dimensions, or even of correlative proportions of locomotives, may be deduced as better than any and every other. For whatever may be the unchanging proportions of a locomotive, it is seldom performing the same amount of work continuously for five minutes together. In almost every work in which the Locomotive Engine has yet been treated, there is the assumption of some necessarily exact ratio between the diameter or capacity of the cylinders and the area of fire-grate, a definite ratio of fire-grate to tube surface, and of one thing to another throughout the whole machine. These are generally taken empirically from some successful engine or class of engines, and are doubtless worth considering; yet, in the course of a year or two from their authoritative deduction, these proportions or ratios are found to be, perhaps, only a little better or a little worse than some others widely differing from them. The amount of fuel which may be burnt does not depend upon the size of a fire-grate, but upon the amount of air which may be brought in contact with the incandescent coke or coal in a given space of time. The evaporative power of a boiler does not depend absolutely upon the extent of its heating surface, but conjointly upon its extent, and the mode of its disposition, and the degree of heat applied. Neither is there any exact proportion of cylinder to boiler, or of stroke to diameter of driving-wheel, or any definite pressure, or any grade of expansive working, which is necessarily better than any and every other. The best locomotives present no more than a judicious compromise between one series of principles which would require that all their dimensions were enlarged, and another series upon which the size and capacity of every part should be diminished. The Locomotive Engine, it may be boldly said, owes nothing to mathematical investigation. In its earliest forms it was designed by unlettered men, and it has grown into what it now is solely through the results of practical experience. That its most useful proportions may be defined within certain limits is most true, but these limits have a much wider range than is commonly admitted. This must of necessity be the case while there is a weighty *pro* and a not less weighty *con* affecting every single detail of the engine. Thus, a large fire-grate facilitates a more thorough, and therefore a more economical, combustion of fuel, and with less injury to the plates forming the sides of the fire-box. On the other hand, it has the disadvantage of increased weight, and it requires greater care on the part of the stoker. This is but one, and possibly not the most conclusive, illustration of the inevitable latitude which must exist in all questions of locomotive proportion; yet the Author would not hesitate, were it worth while to do so, to give a good practical reason why every single part of a locomotive engine should be made larger than it now is, and an equally good contrary reason why it should be made much smaller. Practice, and practice only, has enabled engineers to fix, with some approximation to closeness, upon the golden mean. No necessarily fictitious exactness in these respects will be attempted in the present work, and the book must be left, therefore, to take its chance with those who insist upon the application of a formula to everything material.

A knowledge of Locomotive Engineering may be gained in two ways: either separately or conjoined. One is by direct employment in the construction and working of locomotives; the other is by a close observation of what is done by others, and by reading. There are engineers who, naturally apt, have so far mastered their business in the workshop as to command all the confidence they desire, and there are others who, not less successful in this respect, have never gone beyond the drawing office. But each of these classes of engineers works at an obvious disadvantage; for their profession is one which can be best exercised only with a full knowledge of both its principles and practice. The mere workman dare not trust himself out of the beaten track, and whatever authority he may gain, he will be almost certain to perpetuate any established evil within his sphere; while the mere draughtsman will overlook important practical details, and, if he have a great notion of the scientific principles of his profession, will often readily sacrifice matters of vital consequence in order to carry out some favourite theoretical deduction. It occasionally happens that civil engineers—generally clear-headed and possessing that kind of education which serves best in a systematic study of the Steam Engine, but wanting in practical acquaintance with locomotives—try their hands at "improving" them, and in too many cases with much loss to their clients. We could name more than one railway company, were it necessary to do so, the shareholders in which have, after vexatious and costly litigation, suffered heavy losses in consequence of the absolute impracticability of the plans of locomotives prepared by their civil engineering advisers, and sent out, under peremptory specifications, to locomotive makers.

The best mode of mastering the Locomotive Engine must, we think, ever be a preliminary pupilage in a locomotive factory or railway workshop, and, subsequently, a term of study devoted to the principles involved in the generation and application of steam, and in the construction of railway machinery. There are but few who, having once, as they may choose to believe they have done, mastered what we may describe as the theory of the Locomotive, will afterwards voluntarily submit to the routine (which may to them appear drudgery) of a workshop, and hence the practical branch of the profession must be generally taken up first, if at all. It would of course be best, where that is practicable, that the principles be studied while the practice is being acquired. But the circumstances under which young gentlemen and workmen are employed in the great locomotive factories and railway workshops hardly admit of this; and it may be concluded that, on the whole, the embryo engineer really loses nothing in deferring his studies, as such, until he has left the fitting and erecting shops, perhaps for the last time. A knowledge of mechanical drawing is of great assistance in the study of all kinds of machinery; and this knowledge may be profitably acquired, as far as circumstances admit, in the intervals of active employment. Marked aptitude in this respect is often sufficient to procure admission to the drawing-office, and the opportunities there afforded cannot be too highly prized. If, however, whether here or in the factory, the pupil has no power of looking beyond his immediate employment, he can hardly ever rise above the level at best of the clever workman or the careful draughtsman. But where, to industrious habits and fair mental activity, there is added a love of his profession for its own sake, the young engineer can hardly fail to attain to eminence.

The present work is intended to serve as a text-book to the student of the Locomotive Engine, and as a work of reference to the engineer in actual practice. Of all that may be learned of a practical subject, but a part only can be taught, and still less can be imparted, in a written volume. It is intended, therefore, that the present treatise shall be as exhaustive as reasonable industry can make it. The Locomotive Engine will be treated as if it embodied the only application of the power of steam to useful purposes, and as if many important principles of mechanism found their expression in its action alone. Principles, as such, will be invested with their true dignity; but in applying principles to practice, great quantitative exactness is seldom necessary in the construction of steam engines, and where it is attempted, a want of experience may be generally suspected. There are many who insist upon rules for proportioning machinery; but in the case of locomotive engines no one set of proportions, or of rules for determining them, has yet been found absolutely superior to any other. Successful designers seldom, if ever, confine themselves to arbitrary rules; for with a knowledge of the principles involved in the consideration of any locomotive detail, and with an acquaintance with what has been already successfully done in respect of it, the engineer finds it more satisfactory to make each dimension the subject of an original process of thought, however brief it may be. To adopt any series of rules, except those which one has gradually determined for himself in his own practice, would greatly hamper the mechanical engineer; and if such a course did not of itself disclose existing prejudices, it would be almost certain to create and encourage them. The present volume will contain no rules for proportioning locomotives, but the reasoning affecting each dimension will be given as far as space will permit, and the corresponding dimensions adopted by the best makers will be also given. Indeed, no other course is open to the Author, for from his position he is able to declare that no set of the rules with which so many works on the steam engine abound is adopted by successful engineers, and it would be unlikely that a set laid down for the first time in this work would meet with any especial favour in this respect. In many cases the approximate ratios of certain dimensions to each other may and will be given, but in most of these cases any assumption of necessary exactness in this respect is to be guarded against. These remarks are intended to apply more especially to the determination of the leading dimensions of engines, and of the sizes of their details. There are rules also, pretending to more or less authority, which assign exact ratios between the heating surfaces, grate area, calorimeter, area of blast orifice, area of chimney, &c. As abstractions from successful practice, these ratios possess interest, but they are capable of almost infinite variation. There is a constant tendency to improvement, the direction of which appears evident enough to those who thoroughly understand the Locomotive Engine as it is. Should we ever realize what now appear to be the latent capabilities of the steam engine, its weight and the cost of working it would be greatly diminished, and its proportions would differ widely from those which now prevail. Without entering here upon the consideration of the various means of probable improvement, and which will receive due amplification farther on, it may be mentioned that the mechanical details of the construction of the Locomotive Engine already surpass in importance those considerations of proportion, &c., more directly affecting the consumption of fuel. The coal and coke now burnt upon British railways cost rather more than one million annually, and if the whole cost of fuel could be saved, the gain would correspond to an increased dividend of but two-thirds of one per cent. upon the whole capital of the railways concerned. Locomotive repairs cost more, however, than fuel; and although it is the fact that in many cases a saving of fuel would cause a further saving in repairs, yet it is still true that, apart from all considerations affecting the generation and economical application of steam to the production of mechanical power, there is a wide field for economy in the judicious arrangement and execution of structural details. These are generally overlooked, if not despised, by the *dilettanti* who so readily bring their mathematical powers to bear upon the steam engine; but the *minutiæ* of locomotive mechanism have an important commercial bearing and a singular mechanical interest nevertheless, and will not be lost sight of here.

It is the Author's intention, then, to record, in the present work, the knowledge which a somewhat extensive and reasonably successful practice has afforded him; and, if the advantages which he has enjoyed for the study of the higher principles of Locomotive Engineering have not been lost upon him, he may hope also to bring something of value to bear upon the science of steam locomotion. He need hardly say that no practice, and no deductions from practice, are finite. Locomotives continue to be made as they are, because they have been found to answer in their present form, and because attempts to still further improve them are attended with more or less expense and risk. Locomotive engines are not the property of those who construct or of those who work them; and the responsibility resting both upon locomotive makers and railway locomotive superintendents is such as to forbid any rush departure from what, after full trial, has been proved at least trustworthy. There is reason to believe that were those who are pecuniarily interested in locomotives to assume the heavy cost of an extensive (it would be too much to say, exhaustive) series of experiments, further great improvements would be the result; but the interest of no single railway company would justify the uncertain and in any case great expenditure which such experiments would involve, even if, in conducting them, there were no danger of jeopardizing the safety of railway passengers. Whatever one Company might discover for themselves would be immediately seized upon by their rivals, and appropriated without cost. Nor could a number of railway Companies be expected to associate themselves together for the purpose of still further improving the Locomotive. The chance of the misapplication of any sum subscribed for the purpose would form one ground of objection to such a course; and inasmuch as, so far as the capabilities of the Locomotive Engine are concerned, all railway companies now enjoy equal advantages, it is manifestly better to await its gradual development to the degree of perfection of which many somewhat sanguine, but nevertheless respectable engineers, believe it to be capable. These remarks are thrown out in order that it may be understood that, however carefully this work may represent the present practice in Locomotive Engineering, it is not the Author's belief that this practice is finite, or, indeed, measurably near the limit of ultimate improvement.

He has only to add here the sincere expression of his thanks to the many gentlemen who have placed at his disposal the abundant materials, representative of existing practice, incorporated in this volume.

LOCOMOTIVE ENGINEERING,

AND

THE MECHANISM OF RAILWAYS.

CHAPTER I.

HISTORY OF THE LOCOMOTIVE ENGINE.

THE inestimable influence which the Locomotive Engine has exercised upon the affairs of the last and present generations gives an interest and a certain importance to its history, which is almost co-extensive with that of the Steam Engine itself. From the time when it was first proposed to apply the power of steam to any useful purposes whatever, its application to road carriages appears to have been also contemplated. Indeed, it could not have required remarkable perceptive power, still less the inventive faculty, to see that the motive force which could draw water from a mine could also be made to turn a wheel, the wheel being either that of a mill or of a carriage. After such an idea had once presented itself, it could, too, be readily brought into something like a practicable form; for there existed in the last and preceding centuries a much better knowledge of mechanism than is now generally supposed. Indeed, most of the elements of the modern steam engine were known before the Christian era. Hero, of Alexandria, whose book, *Spiritalia seu Pneumatica*, was written nearly or quite 200 years before Christ (and which has been translated and reproduced in various languages since the discovery of the art of printing), treated of the expansive force of steam, and described, moreover, the slide valve, the spindle valve, and the common clack, and illustrated the application of a metallic piston to a metallic cylinder. Mr. Bennet Woodcroft, now superintendent of the Specification Department of the Great Seal Patent Office, caused an excellent translation of Hero's work to be made a few years ago from the original Greek, and this translation, dedicated to the late Prince Consort, contains diagrams and descriptions of nearly one hundred inventions which would now be classed under the head of "Mechanical Engineering," all so clearly described as to leave no doubt of an advanced knowledge among the ancients of that class of mechanism out of which the steam engine has been structurally compounded.

When the attempts to apply the power of steam took a useful direction, about two centuries ago, locomotive engines were not overlooked. Solomon de Caus, while confined as a madman in the Bicêtre, at Paris, did not forget to include land transport among the many possible applications of steam power; and although in his principal work, written in 1615,[*] no evidence is given of his suggestion of and confidence in steam locomotion, the Marquis of Worcester, who saw De Caus in 1641,[†] was apprised of his schemes under this head, and is known to have adopted the idea of the propulsion of carriages by steam, although no mention of this is made in his *Century of Inventions*. Savery, too, at the end of the seventeenth century, had given some attention to the subject. Neither De Caus, the Marquis of Worcester, nor Savery, however, did anything in the way of reducing their ideas of locomotive carriages to a practical shape. In 1759, too, a suggestion was made to James Watt, by his friend Robison, to the effect that the power of steam could be made to drive land carriages.

The first self-moving land carriage, or locomotive engine, was made, however, by a Frenchman, Nicholas Joseph Cugnot, in 1769; and a larger engine, made upon his plans in 1771, is still preserved in the Conservatoire des Arts et Metiers, at Paris. Cugnot's locomotive was upon the principle of the double-cylinder high-pressure engine proposed by Leupold in his *Theatrum Machinarum*, published at Cassel in 1720. This consisted of two single-acting cylinders (Fig. 1) placed upon a boiler, and having each a steam-tight piston, moved alternately by steam admitted through a four-way cock. Leupold proposed no means for converting the movements of his pistons into a circular motion, although Papin had already sketched an imperfect contrivance for this purpose.

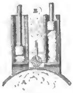
Fig. 1.

Returning now to Cugnot. He was a military engineer, and had commenced a locomotive for common roads in

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[*] *Les Raisons des Forces Mouvantes, avec Diverses Machines tant utiles que Plaisantes*, Paris, 1624. (Dated Heidelberg, 1615.)

[†] In the *Nouvelle Biographie Générale*, vol. ix., p. 258, it is stated that De Caus died about 1635, and that there is no evidence that he was ever confined as a madman. The statement that the Marquis of Worcester saw him in the Bicêtre is contained, however, in a letter from Marion de Lorme to the Marquis de Cinq-Mars, dated Paris, February, 1641.

1769.[*] A Swiss officer, M. Planta, was also occupied with a scheme of his own for the same purpose, and as it appeared that, if successful, such an engine would be of service to the army, Planta's general, Gribeauval, encouraged him, and ordered him also to examine and report upon Cugnot's engine. This, Planta found, was altogether a better contrivance than his own, and he candidly reported in its favour, whereupon, after further examination, the Minister of War, Choiseul, authorized the completion of Cugnot's engine at the cost of the State. On trial it moved at the rate of about 2¼ miles an hour on a level, carrying four persons. The boiler, however, was unable to supply steam for a longer run than a quarter of an hour, at the end of which a stop for about the same length of time became necessary before steam could be raised for another start. The result, however, was deemed sufficiently encouraging to justify a further attempt, and the Government accordingly authorized the construction, at a cost of 20,000 livres, of the locomotive carriage now in the Conservatoire. This was made by Brezin in 1771. It is represented in Figs. 2, 3, 4, and 5, taken from a paper by Mr. E. A. Cowper, read before the Institution of Mechanical Engineers.[†] The engine is upon a large scale, and bears evidence of having been made for work. It has a good-sized copper boiler, poorly contrived, however, for the rapid generation of steam, and it has a pair of 13-inch single-acting cylinders, and a single driving wheel 4 feet 2 inches in diameter and 7 inches broad at the tyre. We speak of this engine in the present tense because it is at this moment fit for service, although its performances would not be likely to compare advantageously with those of modern locomotives, combining arrangements of which it is possible that Cugnot had some idea without having hazarded an experiment upon them. The framing of the engine is in two parts, joined together by a swivelling pin at *e*, exactly over the driving wheel *m*. The hinder framing, *t*, intended for the support of a load of nearly 5 tons, is mounted upon a pair of large wheels, *r r*. The front frame supports the boiler, *c*, and engine. The boiler has a fire-place, grate-bars, and ash-pit at the bottom, and two rectangular flues, serving as chimneys, which rise vertically through the water space. A steam-pipe, *o*, conveys the steam to a pair of single-acting cylinders, *a a*, the admission being effected by a four-way cock, *w*, as in Leupold's engine already described. The pistons in these cylinders have rods working downwards, and connected with pawls, *f*, engaging with a ratchet wheel, *e*, on the driving axle, each stroke of one piston turning the driving wheel one-fourth of one revolution. The two pistons are so connected together that the descent of each causes the ascent of the other, and an effective if not elegant connection is made between the piston-rods and the four-way cock, *w*. The steering gear is exceedingly simple and effective, and the driving apparatus is arranged to work the engine in either direction, the progressive motion being obtained entirely from the adhesion of the driving wheel, *m*, to the ground, the tyre being roughened to increase its "bite." If we take the weight of this engine and its intended load as 12 tons, and its speed as 2¼ miles an hour on a level, it would have about 5 actual horse-power. The resistance at the periphery of the driving wheel may be estimated as 840 lbs., and with a 13-inch stroke of the pistons, this would require a pressure of 19 lbs. per square inch above the atmosphere, and this, therefore, may be taken as about the *datum* upon which Cugnot's calculations were made. The engine, it appears, was tried but two or three times, for, having accidentally overturned in the neighbourhood of which the Madelaine is now the centre, it was immediately locked up in the Arsenal, and all hopes of its ultimate success apparently abandoned.[‡] The engine in its present state bears evidence, however, that even under the most favourable circumstances it could not compete commercially with horse power; and when we recall the generally wretched state of the roads, alike in France and in England, in the last century, we have abundant reason for the discredit which then attached to all schemes for steam locomotion.

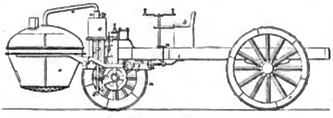

Fig. 2.—Cugnot's Locomotive, 1771.

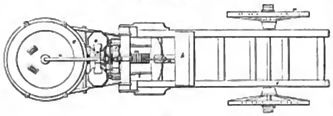

Fig. 3.

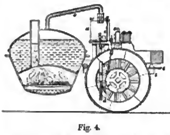

Fig. 4.

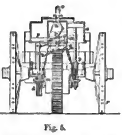

Fig. 5.

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[*] Vide *Comptes Rendus des Séances de l'Academie des Sciences*, 1851, vol. xxxii., p. 524.

[†] *Proceedings, Institution of Mechanical Engineers*, January 26, 1853.

[‡] Arrangements had been made in 1801 to work Cugnot's carriage in the presence of the then General Napoleon Bonaparte. Bonaparte sailed almost immediately, however, for Egypt, and no trial took place.

Although its faults, both of general plan and structural detail, are now most conspicuous, Cugnot's locomotive, judged with fair allowance for the state of mechanical knowledge at the time of its production, is by no means a discreditable work. The obscurity to which it was consigned, however, during the terrors of the French revolution and the turmoil of the first empire, precludes any supposition that this attempt to apply steam to land transport exercised the slightest influence upon the subsequent development of railway locomotion; unless, indeed, some tradition of Cugnot's failure was preserved for the discouragement of those who attempted a like problem in the beginning of the present century. Still, we are not to forget that Cugnot actually made and worked the first locomotive steam engine. He was the first to apply the high-pressure or non-condensing engine, with cylinders and pistons, to the production of rotary motion; and he was the first to apply the energy of steam to what is still the principal sphere of animal power; and he certainly established the fact that steam could be made to serve the purposes of horse traction. That his engine was less economical in its working than horse power, and that it was incomparably inferior to the locomotive of 1863, does not alter the facts that Cugnot—with liberal assistance certainly—took a long stride in advance of his time, and that, as far as he went, his efforts were in the right direction. That his labours were practically lost to society was not his own fault; but, if it be a fault, that of those who should have taken up the subject of steam locomotion where he left it. But, had a locomotive engine, equal to any made at the present time, been produced in the last century, it could hardly have attained anything like commercial success. If intended for railways, it would have long remained in neglect before a company of spirited merchants, like the Liverpool and Manchester proprietors, would have ventured a million or so upon works whose profitable employment was to depend almost wholly upon its success. If intended for common roads, all hopes of its adoption might have been dismissed at once, for common roads a century ago were hardly fitted even for horse-drawn carriages. But not only were there no railways and no good highways, but the foundation of the locomotive engine—the non-condensing high-pressure steam engine—was condemned by every engineer who could then command influence. No man, perhaps, did more in this manner to retard the introduction of the locomotive engine, and therefore the adoption of railways, than James Watt. The Newcomen and Smeaton engines, at the time he commenced to improve upon them, were cumbrous and comparatively ineffective machines, in which steam was condensed within a cylinder and upon one side of a piston, in order that the pressure of the atmosphere might be made to take effect upon the other side. Watt saw the advantage of condensing the steam in a separate vessel, and of entirely excluding the atmosphere from the cylinder, his working pressure being that of steam at or a little above the atmospheric density. Steam of the low pressure required for such a purpose could be raised within boilers of the rudest description. Stone boilers and leaden boilers were common, and as late as 1784 we find Watt proposing a wooden boiler for a road locomotive. Throughout his whole career Watt not only overlooked the advantage of the high-pressure engine, but he resolutely set his face against every attempt to bring it into use. The advantages which Watt's inventions conferred upon society are well known, and it would now be useless to speculate upon what might have been the possible condition of mechanical science and the arts had he never lived; but it is nevertheless almost certain that, had Watt turned his great powers to the improvement of steam boilers, and worked the steam engine at high pressure without condensation, he would have hastened the general adoption of steam power to an extent of which his latest and largest practice could have hardly afforded him a conception. Watt's best engines consumed from 6 to 10 lbs. of coal per horse-power per hour; and in a letter to Mr. Boulton, dated September 23, 1786,[*] he estimated that a locomotive engine must consume 20 lbs. of coal and 2 cubic feet of water per horse-power per hour—a rate which is nearly or quite four times the present ordinary consumption in railway engines; while in some cases an hourly horse-power has been obtained in them (of course without condensation) with less than 3 lbs. of coal. The world must be content to receive great discoveries as they arise, and it is not always that great discoverers obtain their just measure of praise; but, unreasonable and ungenerous as it would be to complain of Watt for not having perfected the use of steam, we cannot forget that he, who did not lack appreciation, uniformly discouraged the efforts of those who sought to improve the steam engine in the most direct manner, by working it at high pressure and at high speed, the advantages of which, for land and marine purposes, are even yet only partly realized in the practice of the present time.

Little, however, as steam locomotion owes to Watt, he patented a locomotive engine. In his patent of 1784 he gives an outline of a steam carriage, and in a letter addressed to Mr. Boulton on the 27th of August of the same year[†] he describes its proposed structure at some length. The boiler was to be "of wood, or of thin metal, strongly secured by hoops or otherwise, to prevent it from bursting by the strength of the steam." There was to be an internal fire box, and, in the letter to Mr. Boulton just referred to, Watt appears to have anticipated the multitubular boiler. "Perhaps," said he, "some means may be hit upon to make the boiler cylindrical, with a number of tubes passing through, like the organ-pipe condenser, whereby it might be thinner and lighter; but," added Watt, who all the while proposed to use steam of hardly more than atmospheric pressure, "I fear this would be too subject to accidents." The "organ-pipe" condenser[‡] was a surface condenser, proposed by Watt in a letter to Dr. Small, and dated 16th March, 1770. The ends of a hollow cylinder 1 foot long were to be drilled to receive a number of pipes of "pinchbeck," only ¼ inch in external diameter and ⅛ inch apart. Condensing water was to be made to pass through these diminutive tubes, and the steam to be condensed was to circulate around them. Whether Watt intended the fire to pass through the tubes of his multitubular boiler, or whether they were to be filled with water, does not appear. To proceed with his locomotive engine: it was to have a single 7-inch cylinder, the piston having a stroke of 1 foot, and working a 4-feet wheel. The piston was to bear a pressure of 6 lbs. per square inch, and was to make 60 double strokes per minute. The whole progressive force of the engine, measured at the periphery of the driving wheels, was to be 80 lbs. only, except in going up hills, when, by means of multiplying gear, and bringing down the speed from 4 miles an hour to less than one, a tractive energy of 400 lbs. was to be obtained. A tubular surface condenser was suggested for this engine, but Watt thought it likely that this could not be depended upon. The engine was to carry but two persons besides the driver, and the whole weight loaded was to be 2,900 lbs., or rather more than 1¼ tons. No engine of this kind was ever made, nor does it appear that its construction was even attempted.

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[*] Muirhead's *Life of Watt*, vol. ii., p. 211, 1854.

[†] Ibid., vol. ii., p. 194-201.

[‡] Ibid., vol. i., p. 99.

William Murdoch, Watt's assistant, took much interest in steam carriages, and, indeed, gave them so much attention, that his master, who after all had no faith in them, had to complain to Mr. Boulton that Murdoch was neglecting the business of the Soho factory. The complaint, it may be supposed, did not fail to reach Murdoch, who contented himself, therefore, with making a toy locomotive, with a single vertical cylinder ¾ inch in diameter, the piston rod, with a 2-inch stroke, being connected to one end of a beam vibrating upon a joint at the other, as in the class of engines now known as "grasshoppers." A connecting rod was jointed to the beam, close to its working end, and turned a crank in the axle of a pair of driving wheels 9½ inches in diameter. The cylinder was half immersed in a small copper boiler, through which a flue passed obliquely, the heat being supplied by a spirit lamp beneath. This miniature locomotive engine, made in 1784, was exhibited by Messrs. James Watt & Co. at the Great Exhibition of 1851, and is still carefully preserved by them at Soho.

William Symington, it should be mentioned, also made a working model of a steam carriage in 1786; but although others were disposed to assist in its introduction, he abandoned the attempt.

Oliver Evans appears to have been the next to give any considerable attention to the practical application of steam to land carriages; but Evans never made a locomotive of any kind, unless the application of wheels to a steam dredging machine, and its propulsion by its own engine from the factory where it was constructed to a river a mile and a half distant, be reckoned tantamount to the production of a locomotive. This performance, however, took place as late as 1804, by which time Trevithick had applied a locomotive engine, with something like success, to the Merthyr Tydvil rail or tramway.

Oliver Evans was an American, and carried on a millwright's business in Philadelphia. The only authentic account of his inventions relating to the steam engine (he was also the father of the self-acting system of machinery now adopted in flour mills in every civilized country) is contained in a small book, entitled *The Young Steam Engineer's Guide*,[*] published in Philadelphia about 1805, although no date appears upon the title page. Evans has occasionally been credited with the invention of the high-pressure engine as early as 1772;[†] but in his book he gives the date of his discovery as 1784, and adds that he only carried it into practical effect in 1801. From the time of his first consideration of the high-pressure engine he appears to have contemplated its application to land transport, and in 1786 he petitioned the State of Pennsylvania for an exclusive right to work his improved flour mills and road locomotives within that State for the term of fourteen years. There was little faith then in steam locomotion, and a patent was granted for the flour mills only. The State of Maryland, however, granted a patent protecting the invention of "steam wagons" for the usual period, and dated 21st May, 1787. Evans states in his book that, in 1794, he entrusted drawings and specifications of his steam engines to a Boston gentleman, Mr. Joseph Stacy Sampson, who was then about visiting England, and who had undertaken to patent his inventions in this country. Mr. Sampson died, however, shortly after his arrival here; "but the papers," wrote Evans, "may have survived." We shall not be justified, however, in devoting much space to Evans; for although his book exhibits a good general knowledge of the mechanical properties of high-pressure steam, and was in many respects far in advance of the prevalent information of the period, he nowhere describes the construction of his proposed steam wagons, nor is it probable that Evans's considerable practice as a maker of high-pressure land engines had the slightest influence upon the subsequent introduction of locomotives in the United States; for, if we are rightly informed, the first locomotive engines employed there were made at Newcastle-upon-Tyne and at Stourbridge.[‡]

It is but fair to add that Evans, in his book, recommended 120 lbs. to the square inch as a good working pressure; that he recommended strong wrought-iron cylindrical boilers 3 feet in diameter, with a single internal flue; that he generally preferred to cut off the steam at one-third of the stroke, leaving it to expand during the remaining two-thirds; and that he worked his engines, according to circumstances, at from 10 to 100 revolutions per minute. His book attests that he had carefully read nearly all that had been done by Watt and his predecessors, and that he was also well indoctrinated with the principles of the steam engine.

We come next to Richard Trevithick, who made and worked the first tramway locomotive. Trevithick, in conjunction with his cousin Vivian (who appears to have been a capitalist rather than an engineer), patented the high-pressure engine and its application to steam carriages in 1802.[*] A steam carriage was made at Vivian's expense, and, after having been worked in Cornwall, was brought up to London and exhibited near the present site of the London and North-Western Railway Terminus at Euston Square. Trevithick was an ingenious man, who occupied himself with inventions apparently as a source of amusement. He tired of exhibiting his steam carriage, and after a few days locked it up, and nothing more was heard of it. It is related that his railway engine was undertaken in consequence of a wager of 1,000 guineas, by Mr. Samuel Homfray of the Pen-y-darren Works, that he would convey a load of iron a distance of 9 miles into Merthyr Tydvil upon a cast-iron tramroad by the power of steam alone. This was accomplished in 1804, but little or nothing was heard of the engine afterwards. It had a single 8-inch cylinder, and a stroke of 4 feet 6 inches. Fig. 6 represents a model locomotive made by Trevithick in 1802, and now preserved in the Museum of Patents, South Kensington. Figs. 7 and 8 have been reduced from an original drawing made by Trevithick, and dated Dec., 1803. It was discovered in Cornwall by Mr. Francis P. Smith, and was brought by him to the South Kensington Museum. It shows the return-flue boiler, and the small chimney, both of incalculable importance in the subsequent development of the locomotive engine.

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[*] Vide *Catalogue* of the Library of the Great Seal Patent Office, Chancery Lane, London. A translation of this work into the French language was published in Paris in 1825, and may be found in the Library of the Institution of Civil Engineers, London. Evans died at New York in 1819, four days after the large works which he had erected in Philadelphia, and upon which he had expended all his means, were burnt to the ground.

[†] *Vide* "Autobiographical Memoir of Evans," *Mechanics' Magazine*, vol. xiv., p. 70.

[‡] These were two single flue engines, one made by Messrs. Robert Stephenson & Co., and delivered from their factory, October 20, 1828 (vide *Transactions of Institution of Civil Engineers* for 1858-59, vol. xviii., p. 52), the other made by Mr. John U. Rastrick at Stourbridge. This engine was mentioned in Mr. Walker's Report to the Directors of the Liverpool and Manchester Railway, March 7, 1829. Both engines were made for the Delaware and Hudson Canal Company, and were started in August, 1829. Both were unsuccessful, and were laid aside after a few weeks' trial.

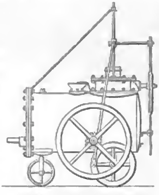
Fig. 6.—Trevithick's Model, 1802.

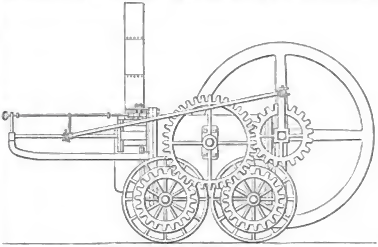
Fig. 7.—Trevithick's Locomotive, 1803.

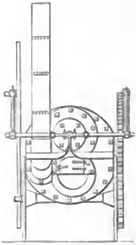
Fig. 8.

In his evidence, given in 1831, before the Committee appointed to inquire into the application of steam to common roads, Trevithick stated that he enclosed the fire in the boiler, and employed a "force draught" created by the steam, for working without a high chimney. The evidence of John Farey,[†] the able historian of the steam engine, and of Goldsworthy Gurney, who from boyhood to manhood knew Trevithick, abundantly established the fact that he turned his exhaust steam into the chimney of his locomotive, although both these gentlemen said that he did not employ it as a "jet" with any effect on the draught upon the fire. That the exhaust steam, however, did, when turned up the chimney, promote the draught was well known in 1805, as will presently be shown. The Merthyr Tydvil engine, too, worked simply by the adhesion of its spur-coupled driving wheels upon the smooth surface of the tramway. This fact has been misstated by one or two authors;[‡] and it has been said that Trevithick did not understand the efficiency and value of the adhesion of a smooth wheel to the surface of a smooth rail, and that he recommended toothed rails, &c., in his specification. The following is the only extract from any of his specifications bearing upon this subject. It is from that of 1802, and was therefore written nearly two years before he had actually tested his locomotive upon a railway. "It is also to be noticed that we do occasionally, or in certain cases, make the external periphery of the wheels uneven by projecting heads of nails, or bolts, or cross grooves, or fittings to railroads, when required, and that in cases of hard pull we cause a lever belt [? bolt] or claw to project through the rim of one or both of the said wheels, so as to take hold of the ground, but that in general the ordinary structure or figure of the external surface of these wheels will be found to answer the intended purpose." The "ordinary surface" was that of a plain cylinder. Trevithick's knowledge that the ordinary adhesion of his driving wheels to smooth rails would occasionally be insufficient was creditable to his proficiency as an engineer, whereas this very knowledge has been misrepresented as ignorance on his part that there was any useful adhesion between two smooth surfaces of iron! His engines were light; and although Mr. Seller had, in 1801,[*] obtained a patent for the coupling rods and right-angled cranks by which the equirotal wheels of all or nearly all locomotive engines are now coupled, Trevithick saw practical difficulties in the way of driving by more than one pair of wheels. Now, at slow speeds, with the full pressure of steam upon the piston (Trevithick employed a single cylinder only, with a great length of stroke), it is as true now as in 1802 that the ordinary adhesion of a single pair of wheels, loaded with but 2 or 3 tons, is nearly useless for any practical purpose on a railway. For there is practical evidence to show (the fact is also borne out by Mr. George Rennie's experiments upon friction[†]) that with very light loads upon driving wheels—say 1 ton—even the ratio or proportion of the friction to the load is less than where the weight is greater. Were we now compelled to resort to weights of from 1 to 3 tons upon the driving wheels of locomotive engines to be worked at slow speeds, means for preventing slipping, even with correspondingly small steam cylinders, would be required.

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[*] Trevithick and Vivian's Specification, March 24, 1802, No. 2,599.

[†] Vide *Parliamentary Inquiry into Steam Carriages*, 1831.

[‡] *Vide* Smiles's *Life of George Stephenson.*

As Trevithick's "force draught" has been mentioned, and an inference drawn from his own evidence that this was the same in effect as the ordinary blast pipe of the modern locomotive, it is now time to commence the history of the steam jet and its offspring the blast pipe.

On the 6th of August, 1805, Davies Giddy, better known as David Gilbert, afterwards President of the Royal Society, wrote to William Nicholson (then editor of *Nicholson's Journal of Philosophy*, better known as the *Philosophical Journal*, and who was subsequently associated with Alexander Tilloch in conducting the *Philosophical Magazine*), stating that he (Giddy) had a short time before examined the working of Trevithick's engine on the Merthyn Tidwell (Merthyr Tydvil) Railway. The waste steam from the single cylinder was discharged upwards into the chimney about 1 foot above its junction with the boiler. When the engine began to move, neither smoke nor visible steam came from the chimney, and even when fresh coal was thrown upon the fire grate, only a faint white cloud made its appearance. Smoke showed itself, however, when the steam was shut off. On looking into the fire place, it was observed by several persons that, while the engine was at work, the fire brightened at each stroke of the piston, and the conclusion was arrived at that the draught was increased by the admission of the waste steam into the chimney. Mr. Giddy's (or Gilbert's) letter, detailing these facts, appeared in *Nicholson's Journal* for September, 1805 (vol. xii., p. 1 and 2). Mr. Nicholson reflected upon the matter, and made an experiment with a small steam jet directed into the mouth of, and thus through, a large open glass tube. This experiment was described by Mr. Nicholson in his *Journal* (vol. xii., p. 47), but he had not then discovered the mode in which the jet acted in causing a draught of air. It was not long afterwards, however, before he ascertained the simple principle upon which the action of the steam jet depends, and he described its properties very clearly in the specification of a patent which he obtained in 1806.[*] Mr. Nicholson did not specifically propose to employ the waste steam of a non-condensing engine for stimulating the fire by which such engine was driven, inasmuch as this had already been done; nor did he place his blast in the chimney, but he did propose to apply high-pressure steam to induce and force along currents of air for various useful purposes, among which was the urging of furnace and other fires. "I convert water," he said, "into steam by the application of heat, in any of the usual and well-known methods, and permit the said steam to rush out through one or more apertures, of such magnitude respectively as may be best adapted for the several intended purposes; and I do permit the current or currents of steam to pass through a portion of the atmosphere, or of air communicating with the atmosphere, or of such other gas or elastic fluid, or vapour, or smoke, as it may be desirable should pass along with the said steam. And, further, I do dispose a tube or pipe (of a circular bore in preference) in such a position, with regard to the said current or currents of steam and air, or of steam and gas, elastic fluid, or vapour, or smoke, as that the said current or currents shall pass through the said tube or pipe, and be carried to its place of destination. And I do give unto the said tube or pipe a greater diameter or size of perforation than I allow for the first escape of the steam from the place where the same was produced; and I do make the said tube or channel either cylindrical, or of such other figure internally as may be best suited to produce that effect which is well known to men of science by the name of the lateral action of fluids, and is described in many books, particularly in some writings of Professor Venturi, which have been inserted in a periodical work called the *Philosophical Journal*, which has been for a number of years published in London, and acknowledged by me as the editor thereof; and the effect of which said lateral action of fluids is particularly to be observed in the ancient and well-known machine for producing a blast by the fall of a shower of water through an upright pipe; and in my said invention the lateral action of the current of steam doth take place with regard to the surrounding air or gas, or elastic fluid or vapour, which is carried along with it, so that the steam is made to produce an effect of the same nature as, but more powerful and advantageous than, is produced by the water in the said ancient machine." Among the various purposes to which the "steam-blasting apparatus" was to be applied, that of blowing fires was included, the application being as follows:—"I do cause the said current of steam and atmospheric air to pass through an aperture or hole into a receptacle or air vessel, wherein the steam is subjected to condensation, and from which vessel the atmospheric air, thus deprived of its steam, is conveyed to a furnace, or any other place containing fire or burning materials, in order that the said current or blast of air shall and may excite and increase the strength, rapidity, and effect of the combustion." Nicholson's specification is illustrated by three diagrams, two of which we engrave. Fig. 9 represents the steam jet issuing from the pipe *a*

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[*] *Vide* Specification of Patent granted to William Seller, 18th June, 1801, No. 2,515, Great Seal Patent Office.

[†] *Philosophical Transactions* for 1829, vol. cxix., p. 159.

[*] *Vide* William Nicholson's Specification, No. 2,990, November 22, 1806, Great Seal Patent Office. *Vide* also *Repertory of Arts* for 1807, vol. x., p. 171, second series.

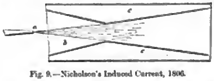

Fig. 9.—Nicholson's Induced Current, 1806.

into the atmosphere, from which air is drawn in and along by the steam, and carried with it into the funnel opening, *b*, of the pipe *b*, *c*, *c*, thence to be directed upon a fire, or for any other purpose. Nicholson also proposed in some cases to surround the pipe *b*, *c*, *c* with steam, or to otherwise expose it to heat, to prevent condensation of the steam passing through. In others he proposed to keep the sides of the pipe cool, on purpose to promote condensation.

Nicholson's next contrivance, shown in Fig. 10, might, had it been followed up, have resulted in the discovery of the injector now used for feeding steam boilers. It is described as "an improvement of the ancient water-blowing machine, in which, in its usual construction, a shower of water carries air down an upright pipe into a regulating belly." Water under pressure was to be introduced through *c* into the close chamber *d*, surrounding the tube *b*, *b*, into which the water enters through apertures, *e*. A jet of steam was to be discharged from the nozzle, *a*, down the axis of the tube *b*, *b*, carrying with it, the inventor appears to have supposed, a considerable quantity of air, while the motion of the steam would be imparted to the water entering at *e*, the action of which would draw in and carry down a further quantity of air in an upright pipe below, as in the old water-blowing machines.

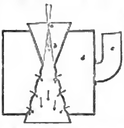

Fig. 10.—Nicholson's Jet Apparatus, 1806.

Nicholson was in intimate communication with a great number of the most distinguished men of science of his time, and this circumstance, and the appearance of his specification *in extenso* in the *Repertory of Arts*, affords reason for supposing that the action of the steam jet became tolerably well known more than half a century ago. In its abstract form it could hardly have failed to come under the notice of those who then and shortly afterwards sought to apply high-pressure steam to locomotion, although any use of the "blast pipe," even as now arranged in the chimney, would appear to have been covered by Nicholson's patent up to nearly the end of the year 1820.

Recurring again to Trevithick's engine of 1804, it appears that, although it performed one or two trips over a rough tramroad at the rate of 5 miles an hour, it was regarded as a commercial failure. Yet, in the hope of something better growing out of it, Mr. Blackett, the proprietor of the Wylam Colliery, near Newcastle-on-Tyne, wrote to Trevithick in Cornwall, in 1809, with reference to a new locomotive. Trevithick replied that he was engaged in other pursuits, and could render no assistance.* It appears that the application was renewed, and that Trevithick reconsidered his decision, for in 1811† one of his engines was sent to Newcastle, although, instead of being set to work upon the Wylam Railway, it was laid aside until some time afterwards, when it was set to drive an iron foundry. It was doubtless a very primitive machine, for George Stephenson, who had never before seen a locomotive, at once declared his belief, after he had examined it, that he could make a better one.

In the same year, 1811, a locomotive engine was commenced by Matthew Murray, of Leeds, an ingenious engineer, who a few years before had patented the short slide valve.‡ This engine, which was first worked in August, 1812, was made for Mr. John Blenkinsop, the proprietor of Middleton Colliery, near Leeds, who had taken out a patent§ for increasing the adhesion of the locomotive by gearing it, by means of a pinion, into a stout rack rail along one side of the line of railway. Murray's engines were made to work upon such a rack, and were thus successfully employed for many years; the Grand Duke Nicholas, afterwards Emperor of Russia, having witnessed their working in 1816; while Tredgold, in his book *On Railroads*, published in 1825, described them as still in use; and in the second edition of Wood *On Railroads*, published in 1831, Blenkinsop's engines are spoken of as having been at work ever since 1811. The construction of these engines is shown in Fig. 11.

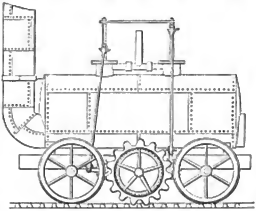

Fig. 11.—Murray's Locomotive upon Blenkinsop's Railway, 1812.

The boiler was cylindrical, with slightly convex ends, and had a single internal flue, in which the fire grate was placed at one end, and out of which the chimney rose at the other. There were two double-acting steam cylinders, the pistons being connected to cranks which were made to work at right angles to each other. This arrangement, an equivalent to which is almost universally adopted in modern locomotives, was introduced by Mr. Murray. The cylinders were upright, and were each immersed for half their length within the boiler, as in Trevithick's engine. Each piston rod was so connected with a pair of cranks, one on each side of the engine, as to drive a toothed wheel, the two wheels thus driven being made to gear into another of twice their diameter, and which maintained the proper position of the cranks with relation to each other. On the axis of the larger toothed wheel just described was a large pinion with coarse rounded teeth, which worked in the rack laid along the railway. The engine was supported upon four smooth wheels, in the manner of an ordinary railway wagon.

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* Vide *Who Invented the Locomotive Engine?* by O. D. Hedley, London, 1858.

† Wood *On Railroads*, first edition, p. 130.

‡ Specification of Matthew Murray's patent, No. 2,632, June 28, 1802.

§ Blenkinsop's Specification, No. 3,431, April 10, 1811.

We have not the precise dimensions of this locomotive; but, with two cylinders, a long stroke, and with the motion of the cranks geared down to the main driving pinion, one turn of which required two double strokes of each piston, it is evident that, with steam of 40 lbs. or 50 lbs. pressure, the tractive force would be considerable, while the speed would be very moderate. The progressive force, measured at the pitch line of the driving pinion, or at the surface of the rail, would probably be considerably beyond the adhesion due to the weight of the engine, which could hardly have exceeded 4 or 5 tons. The railways then in use were laid with cast-iron rails, weighing generally but 36 lbs. per yard, and heavy locomotives were inadmissible. Under the circumstances, then, Murray's engine would, but for Blenkinsop's toothed rack, have wasted a great part of its power in slipping its wheels upon a smooth rail. The rack rail was no doubt a clumsy affair, and the necessity for its use disappeared as soon as heavier engines were adopted, and the practice of gearing down long-stroke pistons to small driving wheels abandoned; but, referring to the actual proportions of the engine, the employment of the rack by no means proves Blenkinsop to have been ignorant of the considerable adhesion always existing between a smooth loaded wheel and a smooth rail. But for the rack rail the Leeds engine could hardly have hauled, as it did, thirty loaded coal wagons, weighing 94 tons, 3½ miles an hour on a level, and 15 tons up a gradient of 1 in 15.[*] Blenkinsop's contrivance has, in later times, been the subject of some ridicule on the part of those who possibly, have not comprehended the nature of locomotive traction so well as Blenkinsop himself. It is certain that, were we now restricted to engines of 5 tons weight, and were it necessary to draw, at the slow speed of only 3 miles an hour, the utmost load for which such engines could supply steam at that speed, we should be compelled again to adopt the rack-rail, or something answering the same purpose.[†]

While Murray's engines, the first ever applied permanently and on a commercial scale to the purposes of railways, were working between Middleton and Leeds, Mr. Blackett, of Wylam, who has been before mentioned, was seeking to make something of the Trevithick engine which he had purchased in the previous year. Mr. Blackett's colliery viewer, William Hedley, was the chief mechanical engineer in this attempt. Hedley first endeavoured to ascertain whether the ordinary adhesion of the wheels of an engine upon smooth rails would be sufficient to insure the useful application of the whole of the steam tractive force. He experimented with wagons driven by manual power, exerted through cranks and gearing, and he subsequently patented[*] a mode of supporting steam engines upon railway carriages. This patent has been said to cover the use of smooth driving wheels for locomotive engines, and it has been claimed for Hedley that he was the father of the present system of working locomotive engines by means of the friction between their own wheels and smooth rails. Although Trevithick had already done this, Hedley, acting for Mr. Blackett, was no doubt the first to work up the adhesion of the driving wheels upon a commercial scale. From what has been said of Blenkinsop it will appear, however, that no discovery was required in order to do this, unless the determination of the extent of the corresponding sacrifice of tractive force at slow speeds was a discovery. In Hedley's specification there is no direct allusion to the employment of smooth wheels. He says (no drawing is given), "I make use of wheels for the carriage, of cast iron or wood, or partly made of wood, by way of increasing the friction or adhesion between the wheels of the conducting carriage and railway; and

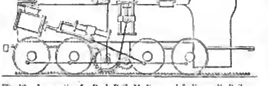

Fig. 12.—Locomotive for Rack Rail, Madison and Indianapolis Railway, U. S., 1848.

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[*] In Mr. Rastrick's Report to the Directors of the Liverpool and Manchester Railway, dated March 7, 1829, he stated that one of Blenkinsop's engines, only a few weeks before, had drawn thirty-eight loaded wagons, weighing 140 tons, at the rate of from 2 to 3½ miles an hour. This was partly upon a level and partly upon gradients descending only 1 in 440

[†] As late as 1848 the rack rail was adopted upon an incline of 1 in 17, on the Madison and Indianapolis Railway, U. S. This incline, which has since been superseded by a line with practicable gradients, rose from the north bank of the river Ohio at Madison, and was about five-eighths of a mile in length. Two engines, called the "John Bright" and the "John Brough," were made by Mr. Baldwin, of Philadelphia, for working this part of the line. Each engine had eight coupled wheels, 4 feet in diameter; and a pair of outside inclined cylinders 17 inches in diameter, and with a 22-inch stroke of piston, were provided for working these wheels only in those parts of the line which were level or nearly so. The rack rail, laid down midway between the ordinary rails of the line, was 10 inches wide, and the teeth had a pitch of 4 inches. The engine had a second pair of 17-inch cylinders, with 18-inch stroke of pistons, placed vertically over the boiler at about the middle of its length. The piston rods worked downwards, and were connected to cranks upon the axis of a stout pinion beneath the boiler. This pinion worked into a wheel of about twice its own diameter, the axis of this wheel being supported in bearings in the lower ends of a pair of connecting rods, which, through the intervention of a bent lever, acted upon by the piston in a horizontal steam cylinder 7 inches in diameter, might be raised or lowered according to the necessity of engaging or disengaging the rack pinion with or from the rack rail. Fig. 12 gives an outline of this arrangement. Each engine had five steam cylinders in all; the boiler had 112 tubes 2 inches in diameter, and of the great length of 17 feet; and the whole weight, exclusive of tender, was upwards of 30 tons. These engines worked for some years with much effect, and with a fair degree of economy.

[*] Specification of Hedley's patent, No. 3,666, March 13, 1813.

where the rail is of cast iron, with a round or flat surface, I attach or cause teeth or flanges to project from both sides of the wheels of the conducting carriage, to enter the ground between the stones, sleepers, or rails, so as to prevent the whole from surging or turning on their own axis." And again the patentee observes, "I also occasionally, in ascending any uneven surface, fix at certain given distances posts or anything to hold by, and from thence extend a rope or chain to my conducting carriage containing the engine, and on the said carriage place a rope roll worked by the toothed wheels mentioned before, to impel the carriage forward." Hedley, therefore, equally with Trevithick and Blenkinsop, understood the insufficiency of the adhesion of light engines when worked at full steam power and at slow speeds. For various reasons, however, he adopted smooth wheels working on smooth rails. One of Blenkinsop's engines, which had been sent to work the railway of the Coxlodge Colliery, near Newcastle, had torn up the rails; the expense of laying down a strong rack along the 5 miles of the Wylam Railway would have then amounted to about £8,000; and there was also Blenkinsop's patent in the way. Hedley, therefore, constructed an engine much upon Trevithick's plan, and which was set to work early in 1813. It had a cast-iron boiler, with a single internal flue, a single cylinder six inches in diameter, and a fly wheel. The engine worked by the adhesion of its driving wheels upon smooth rails. The boiler was deficient in steaming or evaporating power, and much inconvenience was felt from the want of a second cylinder. The results were such, however, as to encourage Mr. Blackett to commission Hedley to make another and better engine (shown in Fig. 13, which also includes a few later improvements). This was at once set about and soon finished. It had a wrought-iron boiler with a return flue, the chimney being placed at the same end as the fire door. Two vertical cylinders were employed, the piston rods being connected to beams centred each at one end—an arrangement known as the "grasshopper." There were four wheels of equal diameter, to which the motion of the pistons was communicated by toothed gearing. The engines subsequently made were of greater power, and the cast-iron plate rails then employed were only able to support them when their weight was distributed upon eight wheels, also geared to work together, so as to obtain the adhesion of them all This class of engines is shown in Fig. 14.

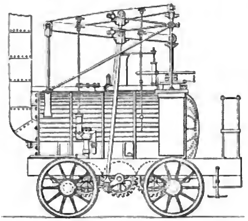

Fig. 13.—Hedley's Locomotive, "Puffing Billy" (now in the South Kensington Museum), 1813.

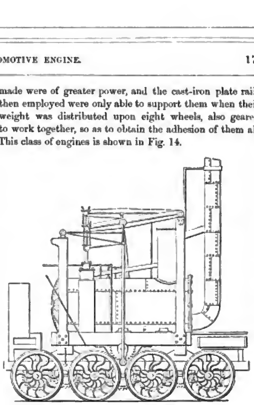

Fig. 14.—Hedley's Engine, Wylam Railway, 1815.

Apart from the use of smooth driving wheels, Hedley's second engine embodied two improvements of much importance. Trevithick was, perhaps, the author of both these improvements, judging from his original drawing of 1803 (Figs. 7 and 8); but it is not certain that Hedley was not the first to bring both into actual use. The return-flue boiler not only gave additional heating surface, but the course of the flame was rendered more effective upon a given area, as a square foot. Although it was not until 1827, or fourteen years afterwards, that Stephenson and Hackworth adopted this form of boiler, it was not greatly inferior to the multitubular arrangement; and, indeed, the "Sanspariel," with the return-flue boiler, was the most effective engine brought forward at the Liverpool and Manchester contest, and but for an accident, in no way attributable to the principle of the engine, would perhaps have won the prize awarded to the "Rocket." Hedley's other improvement was of not less consequence, and consisted in adopting a small diameter, about 1 foot, for the chimney. With the large chimneys previously employed, and those from 20 inches to 22 inches in diameter long subsequently adopted in the Stephenson single-flue engines at Killingworth, the draught was necessarily very sluggish, and the blast pipe, even had it been employed, would have rendered comparatively little assistance. The use of the blast pipe has indeed been claimed for Hedley, but in a letter which he addressed in 1836 to Dr. Lardner, asserting his claim to various improvements in the locomotive engine, he did not mention this. It appears certain, however,* that in Hedley's engines the waste steam was turned up the chimney, although it was first thrown into a cylindrical reservoir, from which it escaped gradually, the object being to keep down the noise otherwise produced. Any admission of high-pressure[*] steam into a chimney will, if directed upwards, produce a draught; and however much the steam may be retarded in reaching the chimney, its final escape therein generates the action attributed to the "blast pipe." This action is that simply referred to in Nicholson's patent of 1806, as the then well-known "lateral action of fluids." A strong jet of water directed through a cistern of water will set its whole contents in motion; and a jet, or even a comparatively gentle puff, of steam will, if discharged into the atmosphere, draw in or "induce" a considerable quantity of air, which will thus acquire the onward motion of the steam itself. There is no difference in kind, but only in degree, between the effect thus produced by a small jet of very high pressure steam and a diffusive exhaust at a pressure only a little above the atmosphere. The distinction has been possibly overdrawn between the simple discharge of the waste steam up the chimney, as practised by Trevithick in 1804 (and to which most locomotive engineers have now returned), and the use of a contracted jet, as first carried out in locomotive steam carriages by Goldsworthy Gurney in 1826.[†] The inductive action of the jet exceeds that of the softened or spent exhaust only in consequence of the greater velocity of the former, the difference in the respective velocities being due wholly to the difference in the pressures generating them. In order to obtain any considerable ejective velocity of exhaust steam a corresponding back pressure must be tolerated in the cylinders, for it is only initial pressure that can give this velocity. Trevithick, or whoever first discharged the exhaust steam from a high-pressure engine up a chimney, could hardly have failed to produce an increased draught, however large the discharging orifice might have been; nor could he have failed to note the corresponding difference in the activity of the fire, when the engine was at work, as compared with what it was when no steam was passing up the chimney. Indeed, one of the most eminent men of science of the day had observed this very circumstance. This and Nicholson's specification justify us in concluding that the action of the exhaust steam in the chimney of a locomotive was well known in 1813, and prepares us to believe the published statement of one of Hedley's sons,[‡] that his father's second engine, made in 1813, had the "blast pipe;" not certainly the contracted orifice subsequently used, for great intensity of draught and consequent evaporative power would have been of no advantage, but rather the contrary, while the engines were still wanting in the weight necessary to give corresponding adhesion.[§] They could not, however well supplied with steam, have been safely driven much faster than 5 miles an hour; and hence, had they been fitted with a contracted blast pipe, the greater quantity of steam produced would have resulted only in increasing the pressure upon the pistons, and in slipping the wheels uselessly upon the rails.

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* Wood *On Railroads*, first edition, 1825.

Almost before the practicability of Hedley's engines had been fully established upon the Wylam Railway, George Stephenson, who had watched their progress from the first, recommended the proprietors of the Killingworth Colliery, where he was employed, to adopt locomotive power in "leading" the coals from the pit. It was finally decided to construct an engine, which was completed under Stephenson's directions in the colliery workshops, and tried for the first time on the 25th July, 1814. It was supported upon four wheels 3 feet in diameter; the boiler, of wrought iron, was 8 feet long and 34 inches in diameter, with a single internal flue 20 inches in diameter, having a fire place at one end, and leading into a 20-inch chimney at the other. There were two vertical cylinders, each 8 inches in the bore, and for a 24-inch stroke. These were immersed for half their length in the boiler, as in Trevithick's engine. The motion of the pistons was communicated to the wheels through cranks and toothed gearing, the arrangement of the latter being almost identical with Hedley's. The axle of each pair of driving wheels had a 24-inch toothed wheel keyed upon it, and, as the axles were 5 feet apart, they were geared together by three intermediate wheels, each 12 inches in diameter. The cranks were made to work at right angles, as represented in Fig. 15, and the pistons necessarily made each two double strokes for each revolution of the driving wheels.

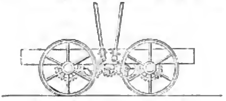

Fig. 15.—Stephenson's Driving Gear, 1814.

The performance of this engine is stated, upon the authority of Mr. Wood, to have been the haulage of eight loaded wagons, weighing 30 tons, up an incline of 1 in 450, and at the rate of 4 miles an hour. The steam pressure is not given; but, making a very liberal allowance for the friction of the engine and train—viz., 17 lbs. per ton—we have a total progressive force of about 800 lbs., corresponding to a pressure of only 9½ lbs. per square inch upon the pistons. Taking the weight of the engine, exclusive of tender, as 6 tons, the adhesion of its coupled wheels upon the rails should have been at least 1 ton, adequate to the draught of at least 100 tons upon an incline of 1 in 450, providing only that an average pressure of 26½ lbs. per square inch was regularly maintained upon the pistons, or say 30 lbs., to allow for the internal friction of the engine itself. Adopting, however, the average pressure of 9½ lbs., which appears to have been the utmost that the boiler could maintain continuously, this corresponds, at 4 miles an hour, to 8½ indicated horse-power. Mr. Smiles states that the average movement of this engine was but 3 miles an hour, and that after a year's trial it was found to have no advantage in point of economy over horse power. If not an entire failure, therefore, it was greatly inferior to the Wylam engines, and even to Murray's, working on Blenkinsop's rack rail.* The radical defects of the Killingworth engine, as, indeed, of all the engines constructed by the same maker down to 1829, were the single flue and the enormous chimney. In the first Killingworth engine, too, the exhaust steam does not appear to have been turned into the chimney at all. The example of the Wylam engines probably led to the subsequent amendment in this respect, for in 1815 Mr. Stephenson endeavoured to obtain the advantage of a steam draught. In Wood's *Treatise on Railroads*, first edition, page 292, this attempt is referred to, and in terms which imply that it was successful; "but," adds Wood, writing in 1825, "the tube through the boiler having been increased (from 20 inches to 22 inches in diameter), there is now no longer any occasion for the action of the steam to assist the motion of the heated air in the chimney." As to the claim which has been put forward for George Stephenson as the discoverer of the blast pipe in 1815, it may be said that he was careful to patent his inventions, and, had the blast pipe been one of these, it would doubtless have been included either in his patent of February 28, 1815, or September 30, 1816.

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[*] The pressure employed by Trevithick, Murray, and Hedley was 50 lbs. per square inch.

[†] *Vide* Report of Lords' Committee *On Accidents in Mines*, 1849, Questions Nos. 3,848 to 3,854.

[‡] *Who Invented the Locomotive Engine?* by O. D. Hedley, London, 1858.

[§] Much controversy was elicited in 1857 in consequence of statements in *The Life of George Stephenson* (by Mr. Smiles) claiming for that engineer the credit of having been the first, in 1815, to adopt the blast pipe. Hedley's second engine, made in 1813, is now preserved in the Patent Museum, South Kensington, and there is affixed to it the following extract of a letter from Mr. Smiles to the late J. F. B. Blackett, M.P., of Newcastle-on-Tyne, and dated September 26, 1854:—

"In the course of certain inquiries which I have been recently making in this neighbourhood, with the object of collecting materials for a biography of the late George Stephenson, I have visited Wylam, and had several interviews with an old servant of yours, Jonathan Foster, respecting the early experiments made by your grandfather upon the locomotive on the Wylam wagonway, and I find from the information which I have obtained, that nearly all the most important improvements which led to its eventual success were worked out at Wylam under the late Mr. Blackett's superintendence. The sufficiency of the smooth wheel and smooth rail for traction, and the steam blast into the chimney, were the most important improvements effected by him; the latter, of great importance and value, has been credited to others, Mr. Blackett's name never having been mentioned in connection with the discovery, from all that I can ascertain."

Hedley is of course entitled, as the engineer, to the credit thus given to Mr. Blackett.

Although toothed gearing, as a means of connection between the pistons and driving wheels, continued to be employed for fifty years at Wylam—viz., from 1813 to 1863—Stephenson soon abandoned it on account of its noise, and because it would not admit of the use of springs. In his second engine, made in 1815,† the vertical cylinders, immersed within the boiler, were retained, but each piston rod was secured to a long cross-head, spanning the whole width of the engine, and having a pair of connecting rods working downwards, and turning one pair of wheels by means of crank pins fixed in their arms. It was proposed to couple the wheels together by a pair of coupling rods working upon cranks forged in both axles, and the adhesion of the wheels of the tender was to be also obtained by coupling them to one pair of the engine wheels by means of an endless chain. Fig. 16 shows the whole arrangement.

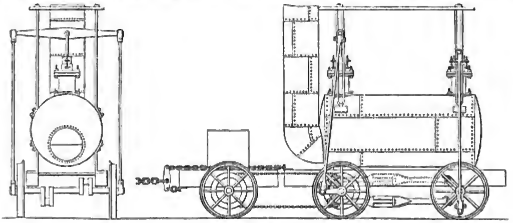
Fig. 16.—Stephenson's Engine, 1815.

The attempt to employ cranked axles was abandoned, and the engine wheels were coupled by an endless chain, until, after some years, outside coupling rods were adopted. The Killingworth engines thus improved, and with the addition of springs (the first use of which is due to Mr. Nicholas Wood), is shown in Fig. 17.

The application of springs to locomotive engines has been mentioned; but it is evident that, with upright cylinders, and with the pistons connected directly to the driving wheels, any vertical motion of the engine would require a corresponding clearance between the pistons and cylinder covers, while also, if the springs were really flexible, the action of the steam upon each piston would, at every stroke, alternately lift and depress the whole weight suspended upon them. George Stephenson sought, therefore, to ease the weight of the engine upon the axles, without at the same time permitting any vertical motion of the boiler other than that of the wheels themselves in rising and falling according to the inequalities of the railway. In his patent of 1816* he describes a mode of supporting a portion of the weight of the engine upon pistons contained in vertical cylinders beneath, and communicating freely with the interior of, the boiler, the pistons being made to press downwards upon the bearings of the axles.

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* In Mr. Rastrick's report to the Directors of the Liverpool and Manchester Railway, March 7, 1829, it was stated that Murray's engines, weighing with tender 6¼ tons, could draw a gross weight of 33¼ tons at the rate of 5 miles an hour at the Middleton Colliery, while Stephenson's engines at Hetton, constructed as late as 1822, and weighing 10½ tons, drew but 36¼ tons at the same speed.

† This was upon Dodd's and Stephenson's patent, of February 28, 1851, No. 3,887.

* Losh and Stephenson's Patent, September 30, 1816, No. 4,067.

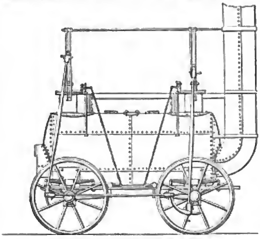

Fig. 17.—Improved Killingworth Engine, 1825.

The drawings attached to the specification show a six-wheeled engine, weighing probably upwards of 8 tons, and having six supporting pistons, each apparently 7 inches in diameter, arranged to be pressed, by the steam in the boiler, downwards upon the journals of the axles. With steam of 50 lbs. pressure per square inch, the whole weight thus supported would be about 5 tons, which approaches the total weight to be supported upon the wheels. The arrangement is shown in Fig 18, representing the pistons supporting part of the weight of the engine. These pistons are open upon their upper surfaces to the water in the boiler. It would be obviously impracticable to exactly balance the pressure upon these six pistons with the weight of that part of the engine supported upon the axles, and indeed, if this were done, the two working pistons would, at each stroke, violently lift and depress the whole machine If, too, the pressure upon the supporting pistons were to exceed the weight to be supported, the boiler and cylinders would be lifted from the axles. The contrivance was introduced into some of the Killingworth and Hetton engines, but it is probable that only about one-half or two-thirds of the weight resting upon the axles was transmitted through the supporting pistons. At any rate they did not move in their cylinders, although they no doubt mitigated the force of the shocks between the engine and the railway, as they allowed only a part of the weight of the boiler and cylinders, instead of the whole, to act percussively upon the axles. In their specification, Losh and Stephenson described the supporting pistons as "floating pistons," which they were not; and they added, evidently without understanding the true action of the pistons, which was different in principle from the action of springs, that, inasmuch as they "acted upon an elastic fluid, they produced the desired effect with much more accuracy than could be obtained by employing the finest springs of steel to suspend the engine." The whole arrangement was, on the contrary, defective in principle, and objectionable on the score of leakage, wear, &c., and, as a matter of course, was ultimately abandoned. The working of the Locomotive Engine was only reconciled with the action of springs by the final adoption of the horizontal cylinder. This had been patented by Symington in 1801,* and was adopted by Trevithick in his steam carriage of 1802, but, although extensively adopted in steam carriages, was only first applied to railway locomotives by Timothy Hackworth, in his engine the "Globe," constructed for the Stockton and Darlington Railway early in 1830.

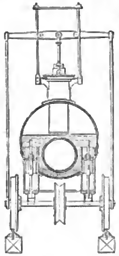

Fig. 18.—Stephenson's "Steam Springs," 1816.

The various arrangements which we have described comprised all that had been done in railway locomotives down to 1827. We should not, however, omit to state that many ingenious men, as Griffiths, Gurney, James, &c., had, between 1821 and 1828, devoted assiduous attention to the introduction of steam carriages on common roads; and if they brought out no sterling improvements, their experiments possessed some value, at least, for railway engineers. Road locomotives had ample adhesion, and only lacked in evaporative power. The steam blast, which for railway engines running at a slow speed and having insufficient adhesion, was as yet hardly required, nor, in some of these engines, even admissible, was therefore of great importance to the road locomotive engineers. Goldsworthy Gurney, then a surgeon and lecturer on chemistry at the Surrey Institution, has claimed the discovery of the properties of the steam jet in 1820—the year, by the way, in which Nicholson's patent for various applications of the steam jet expired. In 1820 and 1821, Gurney applied the jet for promoting combustion in laboratory furnaces, for decomposing various compound bodies, and for working platinum; in 1824 he applied jets of steam, taken directly from the boiler, to increase the draught in the chimneys of the steamboats "Alligator," "Duchess of Clarence," &c.; in 1826 he applied steam in the same manner to increase the draught in the chimney of his road locomotive, and in an *ex parte* statement† not long ago published by Mr. Gurney, he alleges that he supplied Timothy Hackworth with the blast pipe employed by that engineer in the locomotive "Sanspariel," which so nearly won the Liverpool and Manchester prize in 1829.‡ Hackworth, however, appears to have employed the single contracted blast orifice at least two years previously; and with this remark we will now turn to his connection with the Locomotive Engine.

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* Specification of William Symington, October 14, 1801, No. 2,544. The horizontal cylinder was included, also, in the design of Symington's steam carriage, brought out in 1786.

† Mr. Goldsworthy Gurney's *Account of the Invention of the Steam Jet or Blast, and its Application to Steamboats and Locomotive Engines*, London, 1859, pp. 8 and 21.

‡ In the specification of a patent granted March 20, 1828 (No. 5,628), to Nathan Gough, of Salford, for improvements in steam carriages, a single small parallel exhaust pipe is shown with its nozzle directed up the centre of the chimney.

Hackworth was originally employed upon the Wylam Railway, and had, with his own hands, performed much of the work in the construction of the "Puffing Billy." In 1824 he had charge of George Stephenson's locomotive factory at Newcastle, and in June, 1825, he became the locomotive engineer of the Stockton and Darlington Railway. This line was opened for passengers and goods traffic on the 27th of September, 1825, and, in addition to stationary engine and horse power, had five locomotive engines, four of which were from Stephenson's factory. These gave unsatisfactory results, and in 1827 the directors were understood to have seriously considered the propriety of abandoning[*] locomotives altogether, when Hackworth determined to alter and improve one of the engines then in for repairs, and which had been originally made by Wilson,[†] of Newcastle. This engine, when altered, was known as the "Royal George," and commenced working in October, 1827. The boiler was a plain cylinder, 13 feet long and 4 feet 4 inches in diameter. The return flue of the Wylam engines was adopted, and a liberal amount of heating surface thus obtained. There were six coupled wheels, 4 feet in diameter, and the cylinders, which were placed vertically at the end opposite to the fire place, were 11 inches in diameter: the stroke of the pistons being 20 inches, the piston rods worked downward, and were connected to the first pair of wheels. These were without springs, so that the pistons should not jump the engine up and down; but the middle and back pairs of wheels carried their load through stout springs.

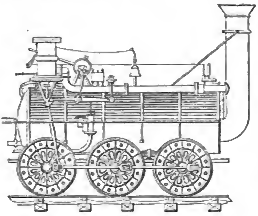

Fig. 19.—Hackworth's "Royal George," 1827.

Sketches of the blast pipe of the "Royal George," and of the exhaust pipes of the Stephenson engines originally employed upon the Stockton and Darlington Railway, were sent by Mr. J. W. Hackworth some time ago to *The Engineer* newspaper, and were published in that journal August 14, 1857 These are reproduced in Figs. 20 and 21.

Separate smoke boxes had not then been provided, and the orifices of the exhaust steam pipes opened into the chimney 4 feet or 5 feet above the bend.[*] In the single flue Stephenson engines, that part of the chimney between the bend and the mouth of the exhaust pipe was often red hot.[†]

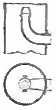

Fig. 20.—Blast Pipe of "Royal George," 1827.

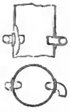

Fig. 21.—Waste Steam Pipes in Chimney of Stephenson's Engines, 1827.

The "Royal George" had a cistern into which a portion of the exhaust steam could be turned to heat the feed water: it had short-stroke force pumps worked by eccentrics, adjustable springs, instead of weights, upon the safety valves, and a single-lever reversing gear. Most, if not all, of these features were novelties first introduced in this engine. Its maximum performance was that of drawing thirty-two loaded wagons,[‡] weighing about 130 tons, at the rate of 5 miles an hour on the level portions of the line, the speed increasing on gentle descents to 9 miles an hour. Its ordinary load in summer was twenty-four loaded wagons, weighing 100 tons; returning with thirty empty wagons, weighing 45 tons,—with which load it surmounted gradients varying from 1 in 450 to 1 in 100. In Messrs. Rastrick and Walker's report to the Directors of the Liverpool and Manchester Railway, March 7, 1829, it was stated, upon the authority of Mr. Robert Stephenson, that this engine had drawn 48¾ tons of goods (exclusive of wagons) for a distance of 2,500 yards, rising 10 feet per mile, and descended again, the average rate of speed over the whole distance, both ways, being 11·2 miles per hour. Messrs. Rastrick and Walker gave the weight of the engine and tender, taken together, as 15 tons, and stated its regular summer load to be 71¼ tons on a level, at 5 miles an hour. Estimating, erroneously no doubt, that the tractive force of the engine was inversely as its speed, they considered it capable of drawing only 28¼ tons regularly at 10 miles an hour. Hackworth's subsequent labours in connection with the Locomotive Engine will be presently mentioned, in describing the Liverpool and Manchester locomotive competition in 1829. This competition resulted in the introduction of the multitubular boiler, the history of which it is now time to commence.

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[*] Even as late as July 7, 1828, Mr. Robert Stephenson found it necessary to write to Mr. Hackworth (and on the 25th of the same month, Mr. George Stephenson also wrote with the same purpose), to ask if it was true that the locomotives of the Stockton and Darlington line were about to be put aside to give place to horses. One of the Stephenson engines had exploded, in consequence of the carelessness of the driver, and the "Royal George" was then under repair. These circumstances possibly explain the cause of the report, prevalent at the time, that the locomotives were about to be abandoned altogether.

[†] This engine had four cylinders, two to each pair of wheels, and appears to have been the first in which a single pair of wheels was worked by two pistons acting upon cranks at right angles to each other. The last named arrangement is one of great consequence in all locomotives, and it is singular that its first adoption has heretofore escaped inquiry. Trevithick's engines had a single cylinder only; Murray's had two, and the pistons were so connected, in the propulsion of the engine, that each should be in the middle of its stroke when the other was at the end of the cylinder. But each piston worked a separate axle, and its effect in helping its fellow piston "over the centres" was necessarily transmitted either through spur gearing, liable to "backlash," or through chains or coupling rods, as in Stephenson's engines. Either of these arrangements was altogether incompatible with any considerable degree of speed; and even in engines working at 4 or 5 miles an hour, both pairs of wheels slipped alternately, at each stroke of the pistons, whenever the coupling chains or rods became a little worn. Wood (vide *Treatise on Railroads*, second edition, 1831, p. 314) states, as the result of his own extensive experiments upon the traction of the colliery locomotives employed down to 1830, that, other things being equal, the available adhesion of the wheels to the rails was one-third less where two pistons were made to work separate pairs of wheels coupled by rods, than where the pistons were connected, through cranks at right angles to each other, to one and the same pair of wheels.

[*] George Stephenson's evidence on the hearing of the Liverpool and Manchester Railway Bill, April 25, 1825. Vide *Blue Book*, p. 206.

[†] Ibid.

[‡] Letter from Timothy Hackworth to Robert Stephenson, March, 1829.

Watt, as we have seen, had, as early as 1784, entertained the idea of a cylindrical boiler with a number of tubes through it, although it is not certain whether he intended to pass the heated air through or around these tubes. The increase of heating surface obtainable with a considerable number of small tubes was well understood, also, from the beginning of the present century; but as this form of boiler was proposed chiefly for generating steam of high pressure, it was naturally sought to make each tube serve as a miniature boiler, which would thus have great strength to resist explosion. In 1803, Arthur Woolf patented an arrangement[*] in which a number of water tubes, to be surrounded by the fire, were employed beneath and in connection with the ordinary cylindrical boiler, the object being simply to increase the heating surface. These boilers were adopted in Cornwall, and one was also set up in Meux's brewery in London, and its construction was described in great detail in the *Philosophical Magazine*. John C. Stevens, of New York, had also obtained an American patent, dated April 11, 1803, for a true water-tube boiler; and he shortly afterwards patented his invention in England.[†] Stevens contemplated very high pressures. In his specification he referred to the experiments upon the temperature and elasticity of steam, carried on in 1790 by M. Betancour, under the auspices of the French Government. These experiments had only been continued up to 280° Fahr.; but, said Stevens, "by experiments which have lately been made by myself, the elasticity of steam at the temperature of boiling oil, which has been estimated at about 600°, was found equal to upwards of forty times the pressure of the atmosphere." For pressures like this, Stevens proposed the following description of boiler, "apprehending," he said, "that the improvement would then be carried to the utmost limit of which the principle was capable." A plate of brass 1 foot square was to be perforated with a number of holes, into each of which would be fixed one end of a copper tube 1 inch in diameter and 2 feet long, the opposite ends being inserted in like manner into a like tube plate. To insure the tightness of the tubes, they were to be "cast" (?) into the plates. A strong cap of iron or brass was to be fastened by bolts to each plate, and so as to leave a water space of an inch or two at each end. Water was to be forced in at one end of the apparatus, which was to be enclosed in brickwork and exposed to fire, and steam was to be taken away from the opposite end, the tubes being placed either vertically or horizontally, as might be preferred. Under a strong fire the water in such a boiler would be quickly forced out of it by the pressure of the steam generated, while under a moderate fire, and at a slow rate of evaporation, the tubes would quickly choke with the impurities ordinarily present in the water. Stevens was a merchant, and had means to conduct costly experiments. In a letter which he addressed, January 12, 1805, to a New York journal called the *Medical Repository*, he stated that he had then been engaged for twenty years, and had spent upwards of £4,000, in experiments upon the steam engine. He had only partially succeeded by purchasing a Boulton and Watt engine, and this he fitted into a steamboat provided with his own water-tube boiler and a screw propeller. This boat was run across the River Hudson, between New York and Hoboken, in 1804. The boiler was 2 feet long, 15 inches wide, and 12 inches high, and contained eighty-one tubes 2 feet long and 1 inch in diameter. Very little steam could be generated, as there were no means of obtaining a draught between the tubes. The inventor proposed, however, to construct a larger boiler for a steamboat to run between New York and Albany, and this boiler was to be 6 feet long, 4 feet wide, and 2 feet deep, the tubes to present 400 square feet of heating surface, upon which the fire was to be made to play by means of an improved draught. These particulars are contained in the appendix to Oliver Evans's *Young Steam Engineer's Guide*, already referred to; but it does not appear that Stevens prosecuted his scheme.

A considerable number and variety of water-tube boilers were subsequently patented, and some of them were brought into partial use. Most of the inventors of steam carriages for common roads, as Griffiths, Gurney, &c., employed water-tube boilers, and Jacob Perkins, who worked steam at such enormous pressures, generated it in closed tubes of small diameter. That the water-tube principle is adapted to locomotive purposes is proved by the fact that the boilers known as Dimpfel's[*] are now exclusively adopted for the coal-burning express engines of at least one important line of railway in the United States,—that between the cities of Philadelphia and Baltimore.

The multitubular boiler, with the heated air passing through tubes surrounded by water, appears to have been first devised by an ingenious London engineer, James Neville, of Shad-Thames.[†] He patented it early in 1826, and shortly afterwards obtained another patent for a steam carriage, in which he proposed to use the same description of boiler. He adopted vertical tubes (see Fig. 22), but observed in his specification that, in some cases, he varied their position and arrangement, placing them in a horizontal or oblique position, particularly where the height of the boiler would be otherwise inconvenient.

Neville stated the object of his improvement to be the generation of a given quantity of steam, by means of a boiler occupying less space than those then in use, and also to economize fuel, by presenting a greater extent of metallic surface, so formed and arranged as to bring the water more closely into contact with the heat. The specification and drawings describe a boiler with an external shell of wrought iron or copper, 3 feet 1 inch in diameter, about 8 feet 9 inches high, and having its upper end formed hemispherically. A cylindrical vessel, 2 feet in diameter and 2 feet 6 inches deep, served as a fire box and close ash pit, a 3-inch water space, in free communication with the sides, extending across the bottom. The top plate of the fire box was pierced with thirty-two holes 2¼ inches in diameter, to receive as many thin copper tubes 4 feet long; and in the centre a tube 6 inches in diameter rose vertically amid the others, all the tubes opening into an enclosed chamber or smoke box in the upper part of the boiler. Three tubes about 3½ inches in diameter descended from this chamber to the smoke box, properly so called, below, whence the products of combustion passed away through an opening at the side. The descending channels, it will be readily seen, were not fully adequate to the discharge of the whole of the heated contents of the rising flues, although the hot air would be considerably reduced in volume by the time it reached the upper ends of the descending flues. A forced blast of air was to be directed into the close ash pit to maintain combustion. The steam drawn from the boiler was to be taken off through a pipe which passed down the centre of the 6-inch flue in the middle of the boiler, the steam being thus superheated, and any water brought over by priming thus vaporized. The enlarged lower end of this copper steam pipe was formed to still further superheat the steam; but it is evident that the fire would act too strongly upon it, and that this part of the apparatus would be after awhile burned out. Neville did not, however, regard the superheater as a necessary part of his plan. A double safety valve was provided at the top of the boiler. The design, as a whole, was meritorious; the boiler would have doubtless answered the purposes of a high-speed locomotive engine; and although a few alterations would have improved it, it contained the whole principle of the locomotive boiler as now made. Neville also proposed, as appears by Newton's *London Journal*, already referred to, to place the tubes in a horizontal or inclined, as well as in a vertical position. He pointed out that the collective area of the flue openings was to be proportioned to the draught, and that not less than 4 square feet of tube surface were to be provided for each horse-power of condensing engines, and not less than 2 square feet per horse-power for high-pressure engines.

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[*] Specification of Arthur Woolf, July 29, 1803, No. 2,726.

[†] Specification of John Cox Stevens, March 31, 1805, No. 2,855.

[*] Patented in England by W. E. Newton, April 17, 1849, No. 12,577. See also *The Engineer*, vol. iv., p. 341.

[†] *Vide* Specification, No. 5,344, March 14, 1826; also Newton's *London Journal of Arts*, vol. xiv., for 1828, p. 295.

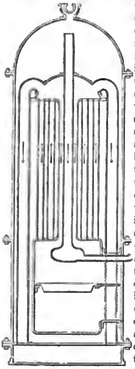

Fig. 22.—Neville's Multitubular Boiler, 1826.

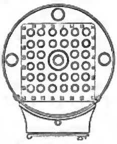

Fig. 23.—Neville's Multitubular Boiler, 1826.

M. Marc Seguin, engineer of the St. Etienne Railway,* near Lyons, patented the multitubular (fire tube) boiler in France, in February, 1828,† and applied it early in the following year to two single-flue locomotives made by George Stephenson. Finding the draught defective, he applied a fan to increase it; but almost immediately afterwards MM. Pelletin and Delabarre, who had re-discovered the steam jet, suggested its application, and an abundance of steam was the result. It does not appear whether the steam blast so used was obtained directly from the boiler, as in Gurney's steam carriage, or from the exhaust steam.

But although all the elements of high-speed locomotive

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* Probably the St. Etienne and Andrezeaux line, of 18 kilomètres or 11 English miles. This line was authorized February 26, 1823, and opened throughout on October 1, 1828. The St. Etienne and Lyons line, of 57 kilomètres, or 35½ English miles, was authorized June 7, 1826. The middle section of 10 miles, from the Rive de Gier to Givors, was opened in October, 1830, while that from Givors to Lyons was only opened in April, 1832, and that from the Rive de Gier to St. Etienne in April, 1833. Vide *Annuaire Officiel des Chemins de Fer*.

† *Vide French Brévets d'Invention*, vol. xxxvii., "Description des Machines." This patent, No. 3,744, was granted to Messrs. Seguin & Co., of Lyons, for ten years, from the 22d of February, 1828. The following is a nearly literal translation of the official description:—

"For an improved construction of a boiler with hollow tubes.

"Steam being produced in proportion to the heating surface, it is believed that the simplest arrangement, occupying the least possible space, is that of a tubular boiler.

"This our invention consists, therefore, as will be seen from the drawing, in the employment of a greater or less number of tubes, through which the heat passes; and these tubes, surrounded by water, present a very extensive heating surface."

Fig. 24 is a copy of Messrs. Seguin & Co.'s drawing, showing the application of the invention to land boilers:—

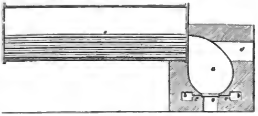

Fig. 24.—Seguin's Multitubular Boiler, 1828.

- *a*, The furnace, "fed by a ventilator or otherwise."
- *b*, Air chambers.
- *c*, Pipes introducing air into the fire place.
- *d*, Door for feeding fuel to the fire place.
- *e*, "Tuyaux calorifères," or heating tubes.
- *o*, Hearth.

In the *Bulletin de la Société d'Encouragement* for 1828, vol. xxvii., p. 424, this boiler is thus noticed in the list of patents granted during the same year:—"Seguin, Annonay, Ardèche, 22me. Fevrier, 10 ans. Chaudière à vapeur sur le principe de l'air chaud circulant dans le tuyaux isolés de petite dimension."

engines were known as early as 1826, high-speed locomotion had not been attempted, nor was it then considered desirable, at least upon railways. It is indeed remarkable, when the relative merits of railways and highways are considered, that the former were commonly regarded as suitable for the conveyance of goods only, while speeds exceeding those of the mail coaches were sought chiefly from steam passenger carriages upon the turnpikes. Indeed, road locomotives were constructed which, had they been made to run upon a good railway, would have doubtless attained a high rate of speed, Gurney's steamer having run at the rate of 14 miles an hour in 1826, as also subsequently, upon the London and Bath road, in July, 1829—a speed exceeding anything up to that time attempted upon a railway.[*] James's steam carriage, also worked at the great pressure of 300 lbs. per square inch, had attained an equally high speed in March, 1829; and although the question of high railway speeds was settled in October, 1829, it deserves to be said that Summers & Ogle's steam carriage, built only a few months afterwards, and upon plans bearing little analogy to those of any railway engines then in existence, attained the great speed of 35 miles an hour upon a common road near Southampton, the steam pressure with a water-tube boiler being 250 lbs. per square inch. The railway locomotives in use in the North down to the year 1829 were, on the contrary, designed and employed only for the conveyance of considerable loads at slow speeds over an inferior description of permanent way. With the exception of Hackworth's engine, they had a stroke of piston of 2 feet, corresponding to the then conventional standard of three times the bore of the cylinder; their wheels were from 3 feet to 4 feet only in diameter, almost always coupled, and in many cases driven by spur gearing; while from the upright position of the cylinders, and the mode of connecting the piston rod to the wheels, springs were inadmissible. Yet Hackworth's and even Hedley's engines, already provided with the steam blast, had an extent of heating surface which, with a shorter stroke and larger wheels, would have given a fair rate of speed—a remark which would have applied also to Stephenson's engines, were it not that they had only the wasteful short single flue adopted at Killingworth in 1814, and with which it was scarcely possible to maintain steam at a speed beyond 8 miles an hour.

It is impossible to believe that the ultimate determination of the Liverpool and Manchester Railway Directors, to adopt locomotive power for working their line, was not in some degree influenced by the considerable success already attained by steam carriages upon common roads. Sir James Anderson and Sir Charles Dance were giving great attention to this class of locomotives. Gurney's engines had been so far improved that, shortly afterwards (in July, 1829), the Duke of Wellington, then Prime Minister, was drawn in his own barouche by one of them at Hounslow; and William Henry James (whose father, a man of wealth and influence, had originally projected the Liverpool and Manchester Railway[*]) had also made and worked a steam carriage with encouraging results. The Liverpool and Manchester Directors had, indeed, previous to March, 1828, authorized their engineer, George Stephenson, to construct a locomotive for their line, and this engine was put to work in the spring of 1829; but being of the very imperfect class supplied by the same maker to the Stockton and Darlington Railway, and the lines near Newcastle, its performances were anything but encouraging in respect of the general adoption of locomotive power. A deputation of the Directors also visited Newcastle and Darlington, and the neighbouring collieries, but their conclusions as to the locomotives employed there were unfavourable. Messrs. Walker and Rastrick, who were engineers of eminence, were finally called in (in January, 1829) to examine the whole subject of fixed and locomotive power; and after careful inquiry, and in the apparent absence of anything better than the Newcastle engines representative of the capabilities of railway locomotives, they reported in favour of stationary engines and rope traction. This was in March, 1829. Messrs. Walker and Rastrick were confirmed in their conclusions by the evidence of Mr. Nicholas Wood, who had seen as much of the Newcastle engines as any man, and who, speaking from his experience with those engines, expressed his decided conviction that no locomotive should be made to go faster than 8 miles an hour.

In the spring of 1829, therefore, the Liverpool and Manchester Directors were upon the point of adopting fixed engines and ropes for working their line. Through the efforts of two or three members of the board it was decided, however, to make a further and conclusive trial of locomotives. George Stephenson, the company's own engineer, was, with the exception of Murray, of Leeds, the only considerable maker of railway locomotives in the kingdom:[†] but as one of his engines had already been tried upon the line, without success, a prize of £500, "open to all comers," was determined upon on the 20th April, 1829, and publicly offered on the 25th of the same month.

The conditions to which the locomotives submitted for trial were required to conform were these:—

1. The engine must effectually consume its own smoke; this condition having been imposed by the Act authorizing the construction of the Liverpool and Manchester Railway.

2. The engine, if of 6 tons weight, must be able to draw after it, day by day, 20 tons weight (including the tender and water tank), at 10 miles an hour, with a pressure of steam on the boiler not exceeding 50 lbs. to the square inch.

3. The boiler must have two safety valves, neither of

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[*] Fair rates of speed were occasionally made upon railways worked by stationary engines and ropes, Mr. Rastrick having reported to the Liverpool and Manchester Directors in March, 1829, that he had a few weeks previously descended one of the planes upon the Brunton and Shields line at a rate varying from 15 to 18 miles an hour.

[*] This line was projected by William James in 1822, and surveyed by Mr. Charles Vignoles in 1823. An agreement is still in existence, dated September 1, 1821, between Messrs. Losh and Stephenson and William James and William H. James, whereby the two last named assigned a part of their patent, for a water-tube boiler, to the former.

[†] In his examination on the Liverpool and Manchester Railway Bill, in April, 1825, George Stephenson stated that he had, up to that time, made fifty-five steam engines, of which sixteen were locomotives. One of his stationary engines was of 200 horse-power. The "Rocket" was the nineteenth locomotive made in the Newcastle factory of Messrs. Stephenson & Co.

which must be fastened down, and one of them be completely out of the control of the engineman.

4. The engine and boiler must be supported on springs and rest on six wheels, the height of the whole not exceeding 15 feet to the top of the chimney.

5. The engine, with water, must not weigh more than 6 tons; but an engine of less weight would be preferred, on its drawing a proportionate load behind it; if of only 4½ tons, then it might be put on only four wheels. The Company to be at liberty to test the boiler, &c., by a pressure of 150 lbs. to the square inch.

6. A mercurial gauge must be affixed to the machine, showing the steam pressure above 45 lbs. to the square inch.

7 The engine must be delivered, complete and ready for trial, at the Liverpool end of the railway, not later than the 1st of October, 1829.

8. The price of the engine must not exceed £550.

To fulfil these conditions nothing more was really required than a shorter stroke and larger wheels than those previously employed; and the combination of these with the return-flue boiler, small chimney, and steam blast of Hackworth's and Hedley's engines. With this very combination, indeed, one of the engines (the "Sanspariel") brought forward as a competitor for the prize took the prescribed load at a mean rate of 15 miles an hour—the greatest rate of speed attained for 1½ miles on a level being 17½ miles an hour.

As for length of stroke, a proportion of three times the bore of the cylinder appears to have been generally considered, forty years ago, as indispensable. If we choose to go back to Cugnot's time, he, it is true, adopted a stroke equal only to the diameter of his cylinders; and Watt, fifteen years afterwards, proposed a 12-inch stroke for a 7-inch piston. But in both Watt's and Cugnot's locomotives four double strokes of one piston were necessary to complete one revolution of the driving wheel. In 1803, a London watchmaker named Freemantle[*] went to the (then) heavy expense of a patent for making the stroke of piston equal to the bore of the cylinder—his object being, however, not an increased number of revolutions, but that of a presumed saving in friction and in condensation, due to external radiation. Hackworth's engine, nevertheless, constructed in 1827, with 11-inch cylinders and a 20-inch stroke, was a bold departure from the practice of the period; and its maker thus led the way in respect of one of the most important proportions of the modern locomotive. As for large driving wheels, Mr. Charles Sylvester, in a professional report to the Directors of the Liverpool and Manchester Railway Company, dated as early as November 30, 1824, recommended a diameter of 5 feet; these, however, with 9-inch cylinders and 24-inch stroke, being intended for a speed of only 9 miles an hour, the wheels making fifty revolutions per minute. A diameter of 6 feet was referred to in the same report as "enormous," but as capable of affording still higher speeds.

On the 1st October, 1829—the day fixed for the competitive trials—four locomotive steam engines were nearly ready; and, to give additional time to the various makers, the contest was postponed until the 6th of the same month. Mr. G. Stephenson entered the "Rocket," in the name of his son, the late Mr. Robert Stephenson; Timothy Hackworth entered the "Sanspariel;" Messrs. John Braithwaite and John Ericsson brought forward the "Novelty;" and Mr. Burstall sent the "Perseverance." The last-named engine was soon found to be unfit to take part in the contest; and, being withdrawn, will not again be mentioned here.

The "Rocket"[*] was a four-wheeled engine, supported on springs, and, with a supply of water in the boiler, weighed 4 tons 5 cwt., exclusive of a tender which, when filled, weighed 3 tons 4 cwt.

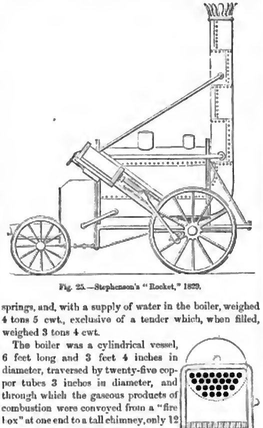
Fig. 25.—Stephenson's "Rocket," 1829.

The boiler was a cylindrical vessel, 6 feet long and 3 feet 4 inches in diameter, traversed by twenty-five copper tubes 3 inches in diameter, and through which the gaseous products of combustion were conveyed from a "fire box" at one end to a tall chimney, only 12 inches in diameter,[†] at the other. The fire grate was 3 feet wide and 2 feet in length in the direction of the boiler, and the fire box, 3 feet deep, and surrounded by a 3-inch water space, presented 20 square feet of heating surface. The water around the fire box communicated with that in the cylindrical boiler through two external pipes. The cylinders, placed in an inclined position, were fastened to the outside of the boiler near the fire box, the connecting rods working upon crank pins in the driving wheels, which were placed under the front end of the engine. The

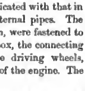
Fig. 26.—Boiler of the "Rocket," 1829.

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[*] *Vide* William Freemantle's specification, November 17, 1803, No. 2,741. This specification also includes a description of slide valve patented only a year before by Murray; and the use of air vessels on the lower or suction sides of pumps. A description and drawing is given of a pump surrounded by an air vessel—an arrangement often patented since.

[*] Now preserved (although altered in some respects) in the Museum of Patents, South Kensington.

[†] Wood *On Railroads*, third edition, 1838, *vide* Tabular List of Engines made by R. Stephenson & Co., p. 352.

cylinders were bored to a diameter of 8 inches, the stroke being 16½ inches. The driving wheels were 4 feet 8½ inches in diameter, that of the trailing wheels being 2 feet 6 inches. The exhaust steam from each cylinder was carried through a pipe of considerable length, and turned upwards into the chimney. The two exhaust orifices were not, however, contracted so as to produce a strong blast. The multitubular boiler was suggested by Mr. Booth, the secretary of the Liverpool and Manchester Railway Company, and who was doubtless unaware of what had already been done by Neville and Seguin. The external surface of the twenty-five tubes was 117¾ square feet; and although this, as a mere matter of area, was much beyond the allowance of heating surface in any locomotive engine previously constructed—at least in England—experiments subsequently made by Mr. Robert Stephenson* and John Dewrance† showed that the heating surface afforded by small tubes was less than one-third as effective, for a given area, as that directly exposed to the fire in the fire box. And inasmuch as flame or active combustion cannot proceed in small tubes, their surface is correspondingly less effective for heating purposes than that of flues 12 inches and upwards in diameter.

The "Sanspariel" was a four-wheeled coupled engine, and, with water in the boiler, weighed 4 tons 15½ cwt., thus exceeding by 5½ cwt. the weight allowed, by the terms of the competition, to be placed upon four wheels. The tender, when charged with coke and water, weighed 3 tons 6 cwt. 3 qrs.

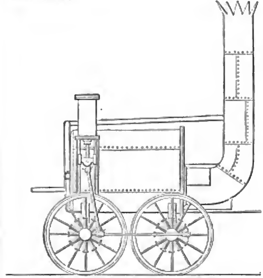

Fig. 27.—Hackworth's "Sanspariel," 1829.

No description (now accessible) of this engine states whether or no it was mounted upon springs. It is improbable that it would have been allowed to compete at all had not the stipulated condition in this respect been complied with; but if springs of any useful elasticity were really employed, it will appear, from the construction of the engine, that a violent up-and-down motion must have been imparted to the boiler at each double stroke of the pistons. The boiler was cylindrical, 4 feet 2 inches in diameter and 6 feet long, and was twice traversed, longitudinally, by a return flue, one length of which, enclosing the fire grate, was 24 inches in diameter, while the other, terminating in the chimney, was 15 inches. Both flues were prolonged 3 feet beyond one end of the boiler, a suitable water space being made around this part of the flues. The chimney was 15 inches in diameter. The fire grate was 5 feet in length, thus presenting 10 square feet of area; that part of the flue answering to an ordinary fire box was 15·7 square feet in extent; while the remaining part of the flue measured 74·6 square feet. The cylinders were vertical, and were 7 inches in diameter, the stroke of the pistons being 18 inches. The connecting rods worked upon crank pins, in one pair of wheels, with which the others were coupled by rods, all the wheels being 4 feet 6 inches in diameter. The exhaust steam from the cylinders was sent through pipes into a single contracted nozzle, or blast pipe, turned upward in the centre of the chimney.

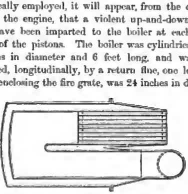

Fig. 28.—Boiler of "Sanspariel," 1829.

The "Novelty" was a four-wheel tank engine, carrying its supply of coke and water upon its own frame. The engine, with water in the boiler and with water and fuel in the tank, weighed 3 tons 17 cwt. 0 qr. 14 lb. As an allowance was necessary for a part of the weight of the framing, wheels, &c. supporting the tank, in order to apportion the load in the ratio adopted for the other engines (the tender being reckoned as a part of the load), the weight of the "Novelty" as an engine only, or irrespective of its carrying capacity for fuel and water, was taken as 2 tons 13 cwts. 2 qrs. 3½ lbs.

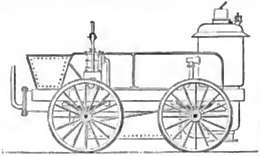

Fig. 29.—Braithwaite and Ericsson's "Novelty," 1829.

The boiler of the "Novelty" comprised an upright cylindrical vessel or fire box, and a horizontal barrel, 15 inches in diameter and about 12 feet long. The upright vessel enclosed a fire box surrounded by water, and having a fire grate 18 inches in diameter. The ash pit was air-tight, the fire being forced by air from a bellows worked by the engine. Coke was fed upon the grate through an air-tight hopper descending in the centre of the upright fire box. The products of combustion were conveyed away through the horizontal barrel by means of a tube folded backwards and forwards, so as to traverse the barrel three times before reaching the chimney. This tube was 4 inches in diameter at the fire box end, and 3 inches in diameter where it united with the chimney, and its total length was about 36 feet. With the exception of this tube, the horizontal barrel was filled with water, the water level being above the enlarged part of the upright vessel. The area of the fire grate was 1·8 square feet, that of the fire box heating surface about 9½ square feet, and that of the tube 33 square feet. The two steam cylinders[*] stood vertically upon the framing, and were 6 inches in diameter, the stroke of the pistons being 12 inches. The piston rods worked through the top covers of the cylinders, crossheads and side rods being employed to communicate the motion, first to bell cranks, whence connecting rods extended to and worked upon cranks in the axle of one pair of wheels. The diameter of each of the wheels was 4 feet 2·1 inches,[†] and coupling chains were provided for, although the two pairs of wheels were not coupled during the trial. The exhaust steam was turned directly into the open air.

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* Wood *On Railroads*, second edition, 1831, p. 403.

† The late Mr. Dewrance was the locomotive superintendent of the Liverpool and Manchester Railway. For an account of his and Mr. Wood's experiments, vide *Three Reports on the Use of the Steam Coals of the Hartley District of Northumberland*, London, 1858, p. 60.

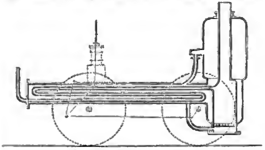

Fig. 30.—Boiler of "Novelty," 1829.

The trials did not finally commence until the 8th of October. The length of run adopted was 1½ miles only, on a level part of the line at Rainhill, near Liverpool. At each end of the run a further distance of ¼th mile was allowed for getting the engines up to their full speed, and for coming to a stop. It was arranged that each engine should traverse the measured distance forty times in all; the first series of ten runs in each direction, making 30 miles, being reckoned equivalent to a journey from Liverpool to Manchester, while the second similar series would correspond to the return journey. The total distance run in each journey, making allowance for the ends of the stage, was 35 miles; but the observations of speed were confined to the distance between the 1½ mile posts. The trials were made under the directions of J. U. Rastrick, Esq., of Stourbridge, Nicholas Wood, Esq., of Killingworth, and John Kennedy, Esq., of Manchester. To each engine was assigned a load (the tender being reckoned as a part of this load) equal to three times its own weight. The manner in which the load of the tank engine "Novelty" was determined has been already described. It was attempted to ascertain the time occupied and quantity of fuel consumed by each engine in raising steam to the prescribed pressure of 50 lbs. per square inch, and to note also the quantity of coke and water consumed on the trip.

The "Rocket" was the first engine tried, and the experiments upon it extended over nearly the whole of the first day. Its own weight, in working trim, being 4 tons 5 cwts., and that of its tender 3 tons 4 cwts. 0 qr. 2 lbs., two wagons laden with stones, and weighing 9 tons 10 cwt. 3 qrs. 26 lbs., were attached to it, making the whole moving weight exactly 17 tons. Steam having been raised from cold water in 57 minutes, the engine entered upon its first journey at 36 minutes 5 seconds past ten in the morning. The first 30 miles, exclusive of startings and stops, or 35 miles in all, were completed at 48 minutes 38 seconds past One.[*] The engine drew its train behind it in going from west to east, and pushed its tender and load before it in returning in the opposite direction. The time occupied in going at full speed, or nearly so, in the twenty to-and-fro runs between the 1½ mile posts, was 2 hours 14 minutes 8 seconds, or an average for the 30 miles of 13·42 miles an hour. The quickest run was in 4 minutes 12 seconds, or at the rate of 21·43 miles an hour. The trips from west to east, during which the engine drew its train, occupied 1 hour 2 minutes 21 seconds, corresponding to 14·43 miles an hour; while the runs from east to west, throughout which the engine pushed its train, occupied 1 hour 11 minutes 47 seconds, corresponding to 12·54 miles an hour. The engine then entered upon its second "journey" of 30 miles,—or, inclusive of the distance ran in starting and stopping, 35 miles. The total time occupied in running at full speed between the 1½ mile posts was 2 hours 6 minutes 49 seconds, corresponding for the 30 miles to 14·2 miles an hour. The trips in which the engine drew its train occupied 57 minutes 12 seconds, corresponding to 15·73 miles an hour, while those in which the tender and wagons were pushed occupied 1 hour 9 minutes 37 seconds, corresponding to 12·93 miles an hour. The quickest run of 1½ miles occupied 3 minutes 44 seconds, corresponding to 24 miles an hour,[†] which was the maximum speed attained on the trial. This was in drawing, and not in pushing the train. Had the train been drawn in every run, the mean speed, which for the 60 miles was 13·8 miles an hour, would probably have reached 15 miles an hour; but it does not follow that a continuous speed at this rate could have been maintained for 30 miles together: for the "Rocket," on the first day of her trial, derived but little benefit from the discharge of the exhaust steam up the chimney, and, indeed, made steam nearly as freely when standing as when running. As each run with steam on occupied less than 7 minutes upon the average, any loss of pressure would be quickly made up in stopping at the ends of the stage. A continuous speed of 13·8 miles an hour, the average rate of full speed in drawing and pushing on forty runs, was probably, therefore, as much as the engine would have accomplished had it drawn its train 30 miles without stopping. The whole load, including the tender, was afterwards taken from the engine, when it ran 7 miles in 14 minutes 14 seconds,* or at the rate of 29½ miles an hour; and a carriage containing 36 passengers was drawn at a rate occasionally reaching to 28 miles an hour. The whole of the experiments with the "Rocket" were performed without accident of any kind, and with no delay beyond that inseparable from the circumstances under which the trials were conducted. This result was undoubtedly due, in a great measure, to the extensive and varied experience which the maker already possessed as a constructor of railway and pit machinery.

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[*] The "Novelty" has been generally described as a single cylinder engine. In Messrs. Stephenson and Locke's report, however, *On the Comparative Merits of Locomotive and Fixed Engines*, February, 1830, p. 66, the cylinders are spoken of in the plural number. *Vide*, also, *Mechanic's Magazine*, vol. xii., p. 142. The author is furthermore assured by Mr. John Braithwaite that there were two cylinders working cranks at right angles to each other.

[†] The diameter of the "Novelty's" wheels has been generally given as 4½ feet. The correct measurement is that of Mr. Charles Vignoles, and is given in the *Mechanic's Magazine*, vol. xii., p. 165.

[*] Mr. Smiles, vide *Life of George Stephenson*, has hastily seized upon these figures—viz., 1 hour 48 minutes 38 seconds—as the *period of time* occupied in making the journey of 35 miles, and inclusive of the time lost in twenty starts and twenty stops. The second series of journeys commenced at 3 minutes 12 seconds past Two, and Mr. Smiles accordingly gives the time occupied in the second series of experiments as 2 hours 3 minutes.

[†] In Wood *On Railroads*, second edition, 1831, p. 371, a speed of 29½ miles an hour is erroneously deduced from a run of 1½ miles in 3 minutes 44 seconds, and this error has been repeated by nearly every subsequent writer. In Messrs. Stephenson and Locke's report, February, 1830, *On Locomotive and Fixed Engines*, p. 72, the maximum speed of 24 miles an hour is, however, correctly given.

Besides the coke consumed in raising steam at the commencement of the experiments, 1,085 lbs. were burnt, and 579 imperial gallons, or 92·6 cubic feet of water, were evaporated, corresponding to 5⅓ lbs. of water evaporated per lb. of coke consumed. The hourly rate of evaporation was 18·24 cubic feet, or 114 gallons.

The "Novelty" was tried on the 10th of October. Its proportionate load was assigned to it in the manner already explained, as 6 tons 17 cwt., and consisted of two wagons loaded with stones, the whole weight in motion being 10 tons 14 cwt. 0 qr. 14 lbs. Only two runs of 1½ miles each were made before some part of the machinery gave way, thus putting a stop to the trials. The speed made on the two runs was 16·07 and 13½ miles an hour respectively, the mean of both being rather more than 14¾ miles an hour. Subsequently on the same day the engine, having been repaired, was again tried. The whole weight of the engine and load was 10 tons 6 cwt. 1 qr., and the highest rate of speed attained for half a mile was 21⅙ miles an hour.

The "Sanspariel" was tried on the 13th of October. The weight of the engine being 4 tons 15½ cwt., the tender and three wagons loaded with stones were attached, their weight being 14 tons 6 cwt. 2 qrs., thus making the total weight of engine and load 19 tons 2 cwt. The engine had only commenced running when one of the cylinders cracked through the bore into the steam port extending along its side, the thickness of the metal having been reduced there, by imperfect moulding and boring, to hardly more than ⅛th inch. The pressure of work upon the small shops of the Stockton and Darlington Railway, under Mr. Hackworth's charge, had compelled him to obtain his cylinders from Messrs. Stephenson's factory at Newcastle; and, for some reason, no less than five were cast before the two with which the engine entered upon its trial were obtained, one of these having, furthermore, to be replaced immediately after the trial was concluded. This failure led to a considerable waste of steam at each stroke of the piston; notwithstanding which, however, the engine was kept at work until—22½ miles having been run at full speed—the feed pump stopped working, thus preventing the supply of water to the boiler necessary to continue the experiment. The mean rate of speed for 22½ miles, exclusive of the ends of the stages, was 13·88 miles an hour, the mean speed in drawing the load being 15½ miles an hour, and, in pushing, 12·4 miles. The first run of 1½ miles was made in 5 minutes 9 seconds, or at the rate of 17·47 miles an hour, which was the highest speed attained on the trial.* The consumption of coal, necessarily increased by the failure already described, and by which, also, a considerable quantity of coal was furthermore shot up the chimney in consequence of the increased energy of the steam blast, was at the rate of 692 lbs. per hour during the trial, the hourly evaporation of water being 150 gallons, or 24 cubic feet. Not only was the blast beyond the requirements of the work, but the fire grate was much too long, and a considerable economy would have resulted from shortening it by 2 feet, leaving that additional length of flue-way in the fire tube. The coupling rods were quite unnecessary also for a total load of only 19⅒th tons on a level. The adhesion of a single pair of wheels supporting nearly 2½ tons would have been more than ample for every purpose of the trial, and the engine would have worked more freely, and thus attained a higher speed, had the wheels been uncoupled. The removal of the coupling rods would have also entitled the engine to a deduction equal to three times their weight from the load.

It is to be observed, that in the trial of the "Rocket," as in that generally of any new engine, the speed gradually increased after a few runs. Thus the "Rocket" occupied 7 minutes 43 seconds in making the first run of 1½ miles, corresponding to 11⅔ miles an hour. On the nineteenth run, the time occupied was 4 minutes 12 seconds, corresponding to 21·43 miles an hour. The first 30 miles were made at the rate of 13·42 miles an hour, while the second 30 miles gave a mean of 14·2 miles an hour. It was only towards the conclusion of the experiment that the maximum speed of 24 miles an hour was attained. But for the failure of its feed pump, the "Sanspariel," it may therefore be supposed, would have gradually attained to a higher rate of speed than it did. As far as it had gone, the mean rate of speed was greater than that of the "Rocket," up to the same stage of the experiment. Thus the "Sanspariel" made its 22½ miles in 1 hour 37 minutes 16 seconds, corresponding to 13·88 miles an hour; while the first 22½ miles made by the "Rocket" occupied 1 hour 45 minutes 24 seconds, corresponding to 12·8 miles only per hour. The real power of the "Sanspariel" is to be estimated by its rate of evaporation, which was one-third greater than that of the "Rocket;" and thus the "Sanspareil," after allowing for its greater weight, was the most powerful engine brought forward for trial. Its wasteful consumption of fuel—each pound evaporating but 2⅓ lbs. of water—was in a great measure due to the exceptional causes already explained, while its final break-down upon the course was not attributable to any defect in the principle of the engine, although it was thereby disqualified from receiving the prize which, had it completed its allotted journey, it might have won, even under the disadvantage of a cracked cylinder.

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* Vide *Quarterly Review*, vol. xlii., p. 394.

* A subsequent run, occupying the interval between 11 hours 20 minutes 47 seconds, and 11 hours 26 minutes 26 seconds, or 5 minutes 39 seconds, is, by a typographical error in Messrs. Stephenson and Locke's *Report on Fixed and Locomotive Engines*, p. 76 (copied also by Wood, vide *Treatise on Railroads*, edition of 1831, p. 376), given as 3 minutes 59 seconds, and from this a maximum rate of speed equal to 22½ miles an hour has been erroneously deduced.

On the 14th of October the "Novelty," which had been repaired, was subjected to further trials. It is stated[*] to have taken a load equal to three times its own weight, and with this to have completed a distance of 20⅓ miles in one hour. A coach containing 45 passengers was also drawn at a rate variously stated at from 24 to 32 miles an hour.[†]

The "Rocket," as the only engine which had completed the stipulated distance, received the prize of £500. The "judges" (so called) did not award the prize themselves, but made a report to the railway directors, who thereupon divided that sum between Mr. Robert Stephenson, as the maker of the engine, and Mr. Booth, as the supposed inventor of the multitubular boiler.[‡] The principle and construction of this boiler are certainly preferable to those of the other engines tried, but the performances of the "Rocket" were nevertheless surpassed by both these engines—for the time, at least, during which they continued to work: and their final failure, as Mr. Wood has candidly recorded, did not arise from any defect in the principles upon which they were constructed. It is a common but erroneous opinion that the adoption of locomotive power upon passenger railways, and the consequent great results to which railways have attained since 1829, are due to the success of the "Rocket." The history of the Rainhill contest does not, however, warrant this conclusion, and it is not indeed improbable that the progress and success of the modern railway system would have been nearly the same had the "Rocket" never existed.

After the trials which determined the award of the directors had been made, the three competing engines were put in the best condition for further service. The two exhaust orifices of the "Rocket" were thrown into one, and so contracted that the exhaust steam produced a powerful blast in the chimney. On the 31st October, this engine drew 40 tons at the rate of 13⅓ miles an hour on a level, and 18 tons at the rate of 8 miles an hour up an incline of 1 in 96. The evaporation of water was 185 gallons, or 29·6 cubic feet per hour. Two other performances,[*] upon the incline of 1 in 96 and 1½ miles in length, consisted in drawing a load of 8 tons 10 cwt. in addition to the weight of the engine, 4 tons 5 cwt., and tender, 2 tons 10 cwt., or, in all, 15 tons 5 cwt., at the rate of 16 miles an hour. With the weight of the tender increased to 3 tons, and a load of 13 tons 7 cwt., or 20 tons 12 cwt. in all, the same plane was ascended at the rate of 16 miles an hour.

The "Sanspareil," although its maker offered to guarantee that it should draw 60 tons behind it on a level at 10 miles an hour,—thus exceeding by more than fourfold the stipulated conditions of the Rainhill competition,—was transferred to the Bolton and Leigh Railway. That line then had two planes,—one 1½ miles long, and rising 1 in 72 (or 73⅓ feet per mile), the other 2,200 yards long, and rising 1 in 30, or 176 feet per mile. Up the first-named plane[†] the "Sanspareil" drew its tender with coke and water, two wagons loaded with iron, and a carriage filled with passengers, the whole load, exclusive of the weight of the engine, being 15 tons, the average speed being 9 miles an hour. On the steeper plane the engine took its tender and a carriage filled with passengers, the weight of this load being 4 tons 15 cwt., and the speed from 9 to 11 miles an hour.

The "Novelty," weighing hardly more than one-half as much as the "Rocket" or "Sanspareil" (including tenders), drew 28½ tons on a level at the rate of 8 miles an hour,[‡] the consumption of coke being 0·55 lbs. only per ton of engine and load moved one mile—a rate hardly more than half that of either of its rivals.

We have dwelt at some length upon the Liverpool and Manchester competition, inasmuch as its results unquestionably fixed general attention upon the mechanical and commercial practicability of high-speed locomotive conveyance. The immediate effect of this competition was to enhance the market value of the Liverpool and Manchester Railway shares to an extent corresponding to £65,000 upon the whole share capital of the company, and to stimulate railway enterprise in every part of the kingdom.

Important, however, as were the results attained at Rainhill, they would be reckoned most imperfect now. On her trial the "Rocket" could not have exerted more—taking one hour with another—than 12 indicated horse-power; that of the "Sanspareil" being rather more. The consumption of coke could not have been far from 18 lbs. or 20 lbs. per hourly horse-power in the first-named engine, while the "Sanspareil," even when put in order, would have burned still more than this. After the alterations made subsequently to the Rainhill trials, the "Rocket" exerted about 20 indicated horse-power, while the "Sanspareil," put to drawing heavier trains at a rather slower rate of speed than was first attempted, appears to have exerted about 18 indicated horse-power. The last-named engine continued to work for several years, and rendered good service.

The Killingworth engines, it may be as well to remember, worked up to but about 10 indicated horse-power at the utmost, and to but about 7 in ordinary work. The consumption of coal was between 30 lbs. and 40 lbs. per indicated horse-power per hour; the evaporation being 3½ lbs. water per pound of coal.

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[*] Vide *Mechanic's Magazine*, vol. xii., p. 165, the editor, Mr. Robertson, having attended the whole series of trials, and carefully noted the results.

[†] Messrs. Stephenson and Locke, in their *Report* already referred to, gave the rate of speed (*vide* p. 74) at from 24 to 28 miles an hour. Mr. Robertson (*Mechanic's Magazine*, vol. xii., p. 165) states that 1½ miles were made in 3 minutes, equal to 30 miles an hour; and the *Quarterly Review*, vol. xlii., p. 394, gives the rate as 32 miles an hour; while, without a load, the engine ran a distance of 27¾ miles in one hour, the speed for a part of the way reaching to 40 miles an hour. The engine ran with great steadiness, so that those upon it could both read and write at the highest rates of speed. So surprising were the performances of this engine that the *Liverpool Mercury*, in summing up the results of the contest, observed that, although under the terms of competition, the directors must award the prize of £500 to the "Rocket," the grand prize of public opinion had already been awarded to the "Novelty."

[‡] *Vide* Booth's *Account of the Liverpool and Manchester Railway.*

[*] *Vide* Messrs. Stephenson and Locke's *Report on Locomotive and Fixed Engines*, p. 24.

[†] Vide *Quarterly Review*, vol. xlii., p. 394.

[‡] Wood *On Railroads*, second edition, 1831, p. 398.

The following table will show, in a connected form, what had been done in locomotive construction down to the end of 1829:—

TABLE OF THE PERFORMANCES OF THE EARLY LOCOMOTIVE ENGINES UPON A LEVEL RAILWAY.

Name of Engine.	Railway.	Date.	Weight, exclusive of Tender.	Diameter of Cylinders.	Length of Stroke.	Diameter of Driving Wheels.	Area of Fire Grate.	Area of Fire Box Surface.	Area of Flue or Tube Surface.	Gross Load Drawn, exclusive of Engine.	Speed in Miles per Hour.	Consumption of Fuel per Hour.	Evaporation of Water per Hour.
			Tons.	Inches.	Inches.	Ft. In.	Sq. Ft.	Sq. Ft.	Sq. Ft.	Tons.		Lbs.	Cub. Ft.
Trevithick's, ...	Merthyr Tydvil,	1804	—	(One) 8	54	—	—	—	—	15	5	—	—
Murray's, ...	Middleton, ...	1811	—	—	—	—	—	—	—	94	3½	—	—
Do., ...	Do., ...	1829	6½	—	—	—	—	—	—	140	3	—	—
Hedley's, ...	Wylam, ...	1813	—	—	—	—	—	—	—	—	—	—	—
Stephenson's, ...	Killingworth, ...	1814	—	(Two) 8	24	3 0	6	10	20	40	5	—	—
Do., ...	Do., ...	1815	7½	,, 8	24	3 0	7	11¼	29¾	50	5	Coal, 293	16
Royal George, ...	Stockton and Darlington, .	1827	11	,, 11	20	4 0	—	—	—	130	5	—	—
Do., ...	Do., ...	1829	,,	,, ,,	,,	,, ,,	—	—	—	72	11	—	—
Rocket, ...	Liverpool and Manchester,	1829	4¼	,, 8	16½	4 8½	6	20	117¾	12¾	13·8	Coke, 214	18¼
Do., altered, ...	Do., ...	,,	,,	,, ,,	,,	,, ,,	,,	,,	,,	40	13·3	—	—
Sanspareil, ...	Do., ...	,,	4¾	,, 7	18	4 6	10	15·7	74·6	14¼	13·8	—	—
Do., ...	Do., ...	,,	,,	,, ,,	,,	,, ,,	,,	,,	,,	60	10	—	—
Novelty, ...	Do., ...	,,	2 7/10	,, 6	12	4 2	1·8	9½	33	8	15	—	—
Do., ...	Do., ...	,,	,,	,, ,,	,,	,, ,,	,,	,,	,,	28½	8	—	—

With regard to economy of fuel, the least consumption per ton of load moved 1 mile, was attained by the engines having the least cross-sectional area of flue, or tubes in proportion to the total heating surface. No particulars can now be obtained as to the coal burned and water evaporated in Hedley's engines; but an examination of the results attained by other engines having boilers of like construction, justifies the presumption that the Wylam locomotives worked with greater economy of fuel than any others made prior to the "Royal George" in 1827. The Stephenson engines then at work had single fire tubes, generally 22 inches in diameter, and presenting 380 square inches of cross-sectional area for 11¼ square feet of fire box surface, and 29¾ square feet of flue surface. From the results of various experiments by Mr. Wood, of Killingworth, and others, it appeared that nearly all the engines of this class consumed upwards of 2 lbs. of coal per ton of load moved 1 mile. The "Sanspareil" presented but 176 square inches of flue opening for 15·7 square feet of fire box and 24·6 square feet of tube surface. The actual quantity of coke burned in this engine in the Rainhill trial was weighed, and amounted to 692 lbs. per hour, corresponding to nearly 3½ lbs. per ton of load moved 1 mile; but, inasmuch as this was the consumption with a burst cylinder, and while a direct communication was being opened at each stroke between the boiler and blast pipe, the result is worthless for any purpose of comparison. With the engine in ordinary working order the consumption of fuel would no doubt have been very much less. The "Rocket" had twenty-five 3-inch tubes, the internal diameter of which may be taken as 2⅝ inches, giving 135¼ square inches of tube opening to 20 square feet of fire box and 117¾ square feet of tube surface. The consumption of coke in the "Rocket" was 1·63 lb. per ton of load (exclusive of engine) drawn 1 mile, showing a moderate gain, in respect of economical working, when compared with the earlier engines by the same maker. The "Novelty," with only 7 square inches of flue opening for 9½ square feet of fire box and 33 square feet of tube surface, burned only 0·613 lb. of coke per ton of load (exclusive of engine) moved 1 mile, and in other trials a still smaller consumption was recorded.

Soon after the conclusion of the trials at Rainhill, the Directors of the Liverpool and Manchester Railway ordered from Messrs. Stephenson a number of engines upon nearly the same principle as the "Rocket." The new engines were larger and heavier, however, some of them having 10-inch and others 11-inch cylinders, with 16-inch stroke and 5-feet driving wheels, the weight being from 6½ tons to 7 tons. These engines had outside cylinders fixed to the fire box, but not so steeply inclined as in the "Rocket," the driving wheels being also in front, or nearly under the chimney. The tubes in the boiler were 2 inches only in diameter, and from 90 to 92 in number. A smoke box was also added. The dimensions and performances of two of this class of outside cylinder engines may be given as follows, the boiler pressure being 50 lbs. per square inch:—

OUTSIDE-CYLINDER ENGINES, LIVERPOOL AND MANCHESTER RAILWAY.

Name of Engine.	Diameter of Cylinder.	Length of Stroke.	Diameter of Driving Wheels.	Diameter of Boiler.	Number of Tubes.	Outside Diameter of Tubes.	Length of Tubes.	Area of Grate.	Area of Fire Box.	Area of Tubes.	Sectional Area of Tubes.	Evaporation of Water per Hour.	Consumption of Coke per Hour.	Coke, per Ton of Load per Mile.	Weight of Train, Exclusive of Engine and Tender.	Average Speed in Miles per Hour.
	Ins.	Ins.	Ft.	Ft.		Ins.	Ft.	Sq. Ft.	Sq. Ft.	Sq. Ft.	Sq. Ins.	Cub. Ft.	Lbs.	Lbs.	Tons.	
Phœnix, ...	11	16	5	3	90	2	6½	6	20	306	216	34·4	323	0·78	41½	10
Arrow, ...	10	16	5	3	92	2	6	6	20	289	221	44·0	286	0·67	33½	12

The differences in the dimensions of the two engines do not account for the considerable difference in their performances and working economy.

The steep inclination of the outside cylinders caused the pistons to lift and depress the engine upon the springs at every double stroke, and at moderately high speeds the unsteadiness thus occasioned was very considerable. Timothy Hackworth appears to have been the first to decide upon the arrangement, so widely adopted afterwards, for securing steadiness at high speeds—to wit, horizontal inside cylinders and a cranked driving axle.[*] He had completed his plans for a new passenger locomotive for the Stockton and Darlington line, and embracing these improvements, as early as March 3, 1830, on which day he laid them before Messrs. Stephenson's firm, with an order to proceed with the construction of the engine. The boiler was of the single-flue variety, the fire grate being placed in the flue, in the front end of which a number of water tubes were fixed diametrically across, and in such order as to deflect the heated products of combustion into a spiral course. These tubes were subsequently removed, as they soon choked with matter deposited from the water. The wheels were four in number, 5 feet in diameter, and coupled. A copper steam dome was placed upon the boiler, and from this feature, which appears to have been original at the time, the engine received its name—the "Globe." The driving axle was double cranked, the cranks being at right angles to each other, and the cylinders were placed horizontally, and side by side. The two eccentrics were loose upon the driving axle, and were reversible by a single lever. The "Globe" was not delivered, however, to the Stockton Company until after Messrs. Stephenson had embodied its arrangement of cylinders and driving axle in the "Planet," the first inside-cylinder engine made by that firm for the Liverpool and Manchester Railway, this engine having left Newcastle September 3, 1830, and been placed upon the line on the 4th of the following month.[*] Before the "Planet" left Newcastle, however, Edward Bury, of Liverpool, had placed an inside-cylinder engine upon the Liverpool and Manchester Railway.[†] This was the "Liverpool," designed by Mr. James Kennedy, then the foreman, and subsequently the partner of Mr. Bury; and who also, as early as 1824, had been a foreman in Stephenson's factory at Newcastle. The "Liverpool" had four coupled wheels of the then great diameter of 6 feet. The cylinders were placed nearly horizontally, and side by side; the boiler contained a number of convoluted flues, and the fire was urged by a bellows worked from beneath the tender—an arrangement previously employed by Stephenson in one or two engines constructed shortly before the appearance of the "Rocket." The "Liverpool" commenced working July 22, 1830,[‡] but shortly afterwards broke its crank axle. By the time a new axle had been made the "Planet" had arrived upon the line.

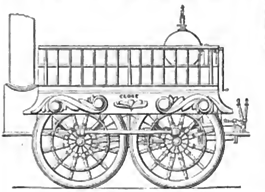

Fig. 31.—Hackworth's "Globe," 1830.

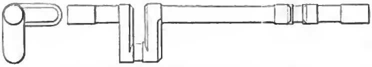

Fig. 32.—Crank Axle of Hackworth's "Globe." Scale, one sixteenth.

In 1830, therefore, three types of horizontal-cylinder locomotives with crank axles were at work—the "Liverpool," the "Planet," and the "Globe." The "Liverpool" was soon afterwards fitted with an ordinary multitubular boiler, and is said to have attained a speed of 58 miles an hour upon a level, with a load of twelve wagons. The "Globe" is stated to have attained a speed of 50 miles an hour, and was constantly employed in the passenger traffic of the Stockton and Darlington line down to 1839, when the boiler exploded at Middlesboro-on-Tees. The "Planet," by Messrs. Stephenson, undoubtedly presented the first combination of the horizontal cylinders and cranked axle *with the multitubular boiler;* and the cylinders were furthermore encased in the smoke box, and thus warmed by the waste heat escaping from the tubes—an arrangement suggested to the late Mr. Robert Stephenson by Richard Trevithick.[§] The constructors of the "Planet," from their established position and long practice in engine making, were enabled to turn to good account the plans and suggestions of Messrs. Hackworth and Kennedy, who had formerly occupied responsible positions in the Newcastle factory, and who still maintained a friendly if not intimate intercourse with their old employers. It must be admitted, to the credit of both the gentlemen just named, as well as to Messrs. Stephenson, that the "Planet" was the prototype of the modern English locomotive, and that for many years it was the model from which both British and American locomotive engineers copied, not only freely, but minutely.[‖] The "Planet" was tried for the first time December 4, 1830, and drew a train of goods and passengers, weighing 76 tons, exclusive of carriages and wagons, from Liverpool to Manchester in 2 hours 39 minutes, the highest speed on a level being 15½ miles an hour. The engine, with coke and water, weighed 9 tons, the tender weighing 4 tons. The cylinders were 11 inches in diameter; the stroke of pistons was 16 inches; the driving wheels were 5 feet in diameter, and the leading wheels 3 feet. The boiler was 3 feet in diameter, and 6½ feet long, the fire box presenting 37½ square feet, and the tubes 370 square feet of heating surface. The tubes were 129 in number, and 1⅝ inches in diameter.

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[*] Vide *Railway Times*, December 20, 1856, containing a letter by John W. Hackworth, detailing the history of the locomotive "Globe."

[*] Vide *Minutes of Proceedings, Institution of Civil Engineers*, vol. xvi., p. 25.

[†] Bury's first engine, on six wheels, commenced working March 12, 1830. This was the "Dreadnought." It was found too heavy, and was soon taken off the line.

[‡] Vide *Minutes of Proceedings, Institution of Civil Engineers*, vol. xvi., p. 22.

[§] Ibid., p. 23.

[‖] The Locks and Canals Company, at Lowell, U. S., commenced the construction of locomotives, after the model of the "Planet," in 1834.

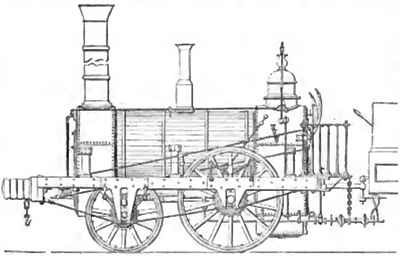

Fig. 33.—Passenger Locomotive by Fenton, Murray, & Jackson, 1834. General type of the "Planet" class.

In the "Planet," then, the Locomotive Engine had assumed a definite and permanent form, compatible with a fair degree of speed and tolerable economy in working. This single engine embodied the results of numberless efforts at locomotive improvement,—for Trevithick's machine of 1804, crude as it was, was nevertheless a locomotive engine, needing only improvement, and in the direction indicated, with tolerable distinctness, by the inventor himself. The comparative success of the "Planet" exacts some acknowledgment of the labours of its constructors, the Messrs. Stephenson. No man contributed so much as the elder Stephenson to the establishment and success of the modern railway system. He knew the advantages, and thoroughly comprehended the economy, of railway transport, before others would admit them. And although it was mechanically and commercially practicable to work railways by means of fixed engines and ropes, Stephenson foresaw the further advantage of locomotive power, and he had an abiding faith in the Locomotive Engine. As improvements appeared he distinguished them clearly, and applied them successfully, in most instances at least; for in the Killingworth, Hetton, and Stockton and Darlington engines he overlooked the advantages of the return-flue boiler and small chimney of the Wylam engine. But there is no ground for asserting, as has been done, that George Stephenson was the inventor of any essential part of the Locomotive Engine; and it is difficult to say in what respect he improved its structure or working, otherwise than by adopting and successfully executing the plans and suggestions of others. There was, indeed, great merit in this,—as much probably as Stephenson ever claimed for himself in this respect; and it no more detracts from his acknowledged sagacity and skill as an engineer, or his singular worth as a man, that he did not "invent" the Locomotive Engine, than it is a reproach to Sir Isaac Newton that he did not originate the electric telegraph. The inventive faculty, indeed, is only rarely found united with that tenacity of purpose, determined energy, and worldly thrift which distinguished the famous north country engineer. There are exceptions to this truth, and of these Stephenson undoubtedly furnished an instance in the invention of the safety lamp. But his earlier projects relative to the Locomotive Engine were, so far as they are disclosed in the specifications of his patents, without practical value. His coupling rods might have been excepted from this remark, were it not that they had been already patented in a better form by Seller. The blast pipe, as has been shown, could not have been Stephenson's, to say nothing of its absence from his specifications of 1815 and 1816, the period to which it has been referred in connection with the second Killingworth engine. The recent ascription of this invention, or rather discovery, to one who, however he may have improved upon the workmanship of Trevithick's engines, rather retrograded in his application of the principles embodied in Trevithick's practice, is the more singular when Giddy's observations, Nicholson's patent, and Venturi's and Dr. Young's researches* are considered. All these had been widely published some years before Stephenson had even seen a locomotive engine; and there is evidence, moreover, that he did not become aware of the true action of the steam jet until it was explained to him by Hackworth at the Rainhill competition in 1829.† The multitubular boiler has never, of

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and supplied many engines of that class to the railways of New England. The first locomotive made at Lowell was "The Patrick" (named for Patrick Tracy Jackson, Esq.), placed upon the Boston and Lowell Railroad, June 24, 1835. This engine, and the "Lowell," "Concord," "Nashua," "Medford," and "Suffolk," on the same line, were measured by the author in 1853; and their dimensions may be given as follows:—11-inch cylinders, 16-inch stroke, 5-feet driving wheels, 3-feet leading wheels; boiler, 2 feet 10 inches in diameter, containing 66 copper tubes, 2¼ inches outside diameter, and 6 feet 10 inches long; fire grate, 1 foot 11 inches long, and 3 feet 6 inches wide; fire box, 3 feet 1½ inch to 3 feet 3 inches deep; grate area, 6·71 square feet; fire box surface (in different engines), from 36·8 to 38·36 square feet; external surface of tubes, 265·6 square feet; induction ports, 6 inches long and 1¾ inch wide; eduction port, 1½ inch wide; two blast pipes, each 1⅛ inch only in diameter at the nozzle; weight of engine in working order, 10 tons 6 cwt. 3 qrs.; weight on driving wheels, 6 tons 9 cwt. 1 qr. 9 lbs.; weight of tender, loaded, 6 tons 13 cwt. 1 qr. 6 lbs. These engines burned wood.

* *Vide* Young's *Natural Philosophy* (1807), vol. ii., p. 534.

† Among the papers of the late Richard Roberts, Esq., is a declaration,

resistance, the speed of an engine will be governed by the time during which the steam is allowed to flow through the induction pipe in the period of each stroke. The longer the time, during these periods, that the steam acts with its full pressure on the piston, the more accelerated will be the speed of the engine, up to the rate at which it can be driven by the steam acting with its full force during the whole duration of every stroke. But it is commonly advantageous that the steam engine should not be constantly wrought to the fullest capability of its power. It is highly important that the same engine should be adaptable by easy means to the performance of very different quantities of work. The most obvious means of effecting this adjustment of power to the variable calls for its employment is by meting out the quantity of motive force delivered to the engine from the boiler. Supposing the steam to continue to be generated of nearly uniform density, the quantity of imparted force will be measured by the length of the intervals of time during which the steam flows uninterruptedly from the boiler into the working cylinder. For the perfect regulation of these intervals of steam supply it is necessary to possess the power of cutting off the steam at any moment of the stroke, leaving the subsequent action to the expansive force of the portion of steam already admitted. My present invention gives the power of adjusting at will, or according to variable circumstances, the moment of cutting off the supply of steam from the boiler, by making one arm of an intervening lever in the working gear of the induction valve changeable at any moment." All this reasoning had probably occurred to Watt half a century earlier; for, according to some accounts, he had devised and applied a mode of regulating the speed of steam engines by a variable cutting off or suppression of steam, according to the work to be done at any moment. Whatever Watt's variable expansion gear was, however, it was not applicable to slide valves, nor, therefore, to locomotive engines. The fact that the action of the slide valve, with respect to the position of the piston, is modified by a change in the travel of the valve, may, and, indeed, should have been known long before 1832; yet, from all that can be gathered, Mr. Roberts was the first to record his knowledge of this fact, and the first to turn it to practical account.

Mr. Roberts, moreover, as a practical engineer, rendered an immediate and important service to locomotive construction. He introduced superior workmanship, giving better proportions and finish to the principal parts, for all of which he made templates, besides referring every important dimension to the standard gauges originally introduced by him in 1825. Not only a man of extraordinary inventive genius,—as the planing machine, the self-acting spinning mule, the slotting machine, wheel-cutting engine, and other invaluable inventions of his sufficiently attest,—Mr. Roberts delighted in refined workmanship in connection with every kind of mechanism. He had served in Maudslay's factory, then the highest school of mechanical engineering in the world; and he was the first who brought to bear upon the modern locomotive that kind of craft which, while it did not alter its general mode of action, nevertheless contributed so much to its working economy. His framing was stronger and the fastenings better. His bearing surfaces were larger and better got up; and in all parts machine-work was substituted to a great extent, in place of the hand-fitting more commonly employed by the north country engineers. Sharp, Roberts, & Co. soon abandoned the construction of locomotives represented in Figs. 38 and 39, and adopted the general arrangement of the Stephenson engines on the Liverpool and Manchester line. To this type, however, Mr. Roberts added characteristic features of design, which still distinguish the English locomotive engine; and by his practice, still perpetuated at the Atlas Works and Gorton Foundry,[*] he sensibly advanced the standard of locomotive construction upon every railway in the kingdom.

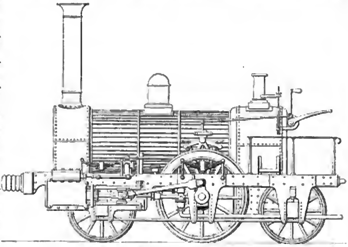

Fig. 42.—Forrester's Locomotive, 1834.

In 1834 Messrs. George Forrester & Co., of Liverpool, completed an engine, of which several copies were afterwards made by the same firm, and which included some important novelties of arrangement, which have been improved upon in subsequent practice. Forrester's engine is represented in Fig. 42. It was of the six-wheel class, and had outside horizontal cylinders. It had, however, outside framing, and the driving wheels were driven by connecting rods acting on cranks keyed upon the axles at a considerable distance beyond the face of the wheels. The centre lines of the cylinders were thus spread widely apart; and, as the wheels were not then counterbalanced, the consequent unsteadiness, at even moderate speeds, was sufficiently apparent, in comparison with the inside cylinder-engines, to gain for those under notice the title of "Boxers." A few pounds of iron properly disposed in the rims of the driving wheels would, however, have redeemed the reputation of these engines. Their outside horizontal cylinders now form one of the best features of modern locomotive practice, and had Forrester employed them in connection with inside framing, he would have anticipated to a remarkable extent the locomotive of 1864. The outside framing, however, permitted the placing of the leading wheels farther forward than would have been otherwise practicable, and there can be no doubt that outside axle bearings are, *per se*, preferable to those inside the wheels.

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[*] Charles F. Beyer, Esq., the principal partner in the firm of Messrs. Beyer, Peacock, & Co., Gorton Foundry, Manchester, was for many years the principal assistant of Mr. Roberts.

The Stockton and Darlington Railway engines, constructed between 1830 and 1834, deserve notice here, not only for their distinctive features, but also for the success with which they worked the coal traffic of the line at that time. These engines, together with Hackworth's "Royal George" of 1827, were the earliest six-coupled engines made, and they invariably had inside framing. Some of the oldest engines still at work upon the Stockton and Darlington line have the return-flue boiler of the "Sanspareil" (see Fig. 28, page 26), a form of boiler first devised by Trevithick. One of these engines, now in use on the South Durham and Lancashire Union line, is represented in Fig. 43, from a photograph taken by John Dixon, Esq., the present venerable chief engineer to the Stockton and Darlington Company. This engine required a tender at each end, although but one is shown. The general disposition of the working parts (shown in Fig. 43) has served as a model for occasional copyists, and one American firm, in particular, continued to employ it, with the addition of improvements, until very recently, for their six-coupled goods engines. The wheels shown were of peculiar structure, and were designed by Timothy Hackworth.

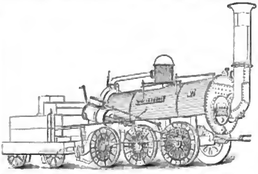
Fig. 43.—Stockton and Darlington Engine, 1830.

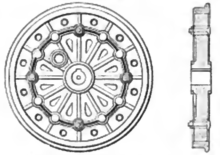
Fig. 44.—Hackworth's Driving Wheels, 1827.

There were no lathes at Shildon in 1830-34 sufficiently large to turn 4-feet wheels when keyed upon the axle. Hackworth, therefore, made his wheels of cast iron, in two portions—to wit, a large central boss and an annular rim. The latter was fastened to the former by oak tree-nails, through which iron bolts or wedges were occasionally driven, to insure greater tightness. The tyres, of wrought iron, were shrunk upon the outer cast-iron rim, which could be readily removed from, or replaced upon, the central boss. These wheels are still preferred for some of the slow-working engines of the Stockton and Darlington line.

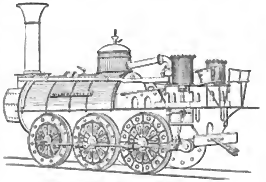
Fig. 45.—Stockton and Darlington Engine, 1832-36.

Fig 45, from a photograph lately taken by Mr. Dixon, represents one of a class of engines made both by Hawthorn and by Stephenson, and in extensive use on the Stockton and Darlington line in 1832-36. These engines had boilers 13 feet long and 44 inches in diameter. The fire place was in a tube 29 inches in diameter and 9 feet long, the fire door being at the same end as the chimney At that end of this tube farthest from the chimney the flame was divided, and the heat returned through a group of copper tubes on each side of the fire place. A smoke box received the products of combustion from both groups of tubes, and communicated with the chimney in the usual manner. There were 53 tubes on each side of the fire tube, or 106 in all; their length being 4 feet and their outside diameter 1¾ inches. The arrangement of the smoke box will be better understood from Fig. 46. These engines required two tenders, one for coal at the chimney end, and another at the opposite end for water.

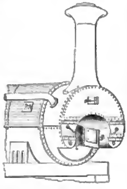
Fig 46.—Smoke Box and Fire Door, Stockton and Darlington Engine.

These engines had generally 14½-inch cylinders, the pistons having a 16-inch stroke. The piston rods worked, as shown in Fig. 45, through the bottom covers of the cylinders, and were connected by short connecting rods to cranks fixed at right angles to each other, upon the ends of a stout shaft revolving in bearings upon the frame.

From this shaft motion was imparted by coupling rods to the three pairs of wheels, each 4 feet in diameter, and mounted on springs. The pressure of steam varied from 36 lbs., in engines considerably worn, to 60 lbs., in those which were new out of the shops. The weight of Hawthorn's engines, made in 1832, was 11¾ tons, and the engines, when worked at moderate speeds, were sufficiently powerful for the heaviest coal trains which the strength of the line and of the wagons would permit of being drawn with advantage.

The early practice of the American locomotive engineers deserves a brief record in the present chapter. As has been already stated, the first railway engine worked in America was constructed in 1828, at Newcastle-on-Tyne. This engine, however, as also its fellow made at Stourbridge, was so imperfect that it was only worked experimentally, when its purchasers determined to abandon for a time the use of steam power upon their railway, a short line owned by the Delaware and Hudson Canal Company. Before referring to the earliest American railway engine, it deserves to be said that a steam carriage for common roads, and made in New York as early as 1829, possessed one feature of importance in connection with locomotive history:—viz., four fixed eccentrics, one for the forward and one for the backing motion of each cylinder. The slide valve of one cylinder had also ½-inch lap at each end; and this cylinder exhausted its steam into the other, although the reason for this procedure is hardly evident. This steam carriage was the work of William T. James, of New York, and the information which the author possesses as to its construction has been communicated to him by Mr. Samuel B. Dougherty, locomotive superintendent of the Camden and Amboy Railroad, connecting the cities of New York and Philadelphia. Mr. Dougherty was the foreman of Mr. James' factory, and he describes the steam carriage in question as having three wheels and a pair of 6-inch cylinders.

The first American railway engine, the "Best Friend," was made at the West Point Foundry,[*] New York, early in 1830. It was designed by Adam Hall, and was sent to South Carolina. Its boiler soon after exploded. No particulars of its construction are now on record. Early in 1831 another engine, the "De Witt Clinton," was constructed at the same establishment, and in July of the same year it was placed on the then Mohawk and Hudson Railroad, now a part of the New York Central Line.

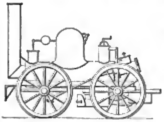

Fig. 47.—American Locomotive, "De Witt Clinton," 1831.

The cylinders were 5½ inches diameter, the stroke of piston being 16 inches. The four wheels were each 4½ feet in diameter. The connecting rods worked upon double cranks in the front axle. The boiler had about thirty tubes of copper, 5 feet long, 4 inches in external diameter, and ¼th inch thick. The weight of this engine was about 4 tons. The particulars of this, and of some of the other early American engines, were communicated to the author in 1858 by David Matthew, one of the oldest locomotive engineers in the States.

A class of engines having much in common with Hedley's "Puffing Billy" was introduced in 1831, upon the Baltimore and Ohio Railroad, the president of the company having, on the 4th January of that year, offered a prize of 4,000 dollars for the best engine of 3½ tons weight, which should draw a load of 15 tons on a level at the rate of 15 miles an hour. The engine was required to have four coupled wheels, and the wheel base was not to exceed 4 feet, to permit of travelling easily around curves of 400 feet (six chains) radius. The maximum limit of pressure was a liberal one—viz., 100 lbs. to the square inch, and the prescribed fuel was anthracite coal. Phineas Davis, of York, Pennsylvania, constructed an engine which more than fulfilled these conditions. The boiler, designed by Peter Cooper,[*] of New York, was upright, with tubes arranged as in Neville's, already described, except that the chimney was at the top, as in all upright tubular boilers now made. The cylinders were placed vertically, and "grasshopper" beams and spur gearing were employed, as in Hedley's engines, to communicate the motion to the wheels. Engines of the same kind, but of greater weight, were for many years employed upon the Baltimore and Ohio Railroad, and with a success for which the high working pressure in some degree accounts.

In July, 1831, Messrs. Robert Stephenson & Co., of Newcastle, sent to the Mohawk and Hudson Railroad an engine—the "Robert Fulton"—having the general features of the "Planet" class, already described, except that, as in others of Messrs. Stephenson's engines made for English lines, the four wheels were of equal diameter (4 feet), and they were coupled by rods working upon pins fixed in cranks upon the outer ends of the axles. Rather more than a year after it was set to work the front wheels of this engine were removed, and a swivelling truck or "bogie" was substituted, this truck having been made upon the model of one already in use under an engine—the "Experiment"—made for the same line at the West Point Foundry, in August, 1832. The "Experiment," constructed by Adam Hall, of New York, was the first to which the swivelling truck or "bogie" was ever applied; this application having been made under the direction of John B. Jervis, the then engineer to the Mohawk and Hudson Railroad, and afterwards the chief engineer to the Croton Aqueduct. The "Experiment" had 9½-inch cylinders, 16-inch stroke, 5-feet driving wheels, a truck of four smaller wheels, and, being intended to burn anthracite coal, the fire grate was 5 feet long. The whole weight of the engine was 7 tons.[†]

Between 1831 and 1836 many English locomotives were sent to the States, where the author has met, even within the last ten or twelve years, with some of the earlier engines by Stephenson, Bury, Forrester, and Braithwaite and Ericsson. The English engines served as models from which the American makers copied largely,—in the case of the Locks and Canals Company (see page 32) minutely,—but it must be owned that, in certain instances, the Americans added original improvements of their own. These were the "bogie," the use of four fixed eccentrics, the spark arrester, separate expansion valves, compensating levers between the springs of coupled engines, &c. The author has it, too, upon credible evidence, that the link motion was applied in almost, if not quite, its present form, in the second engine made by William T. James, of New York, as early as 1832. Mr. Dougherty, who assisted in the construction of this engine, has supplied very full particulars of its dimensions and arrangement, but unfortunately no drawing of it exists, and the only contemporary record of it is contained in the *American Railroad Journal* for 1832, and no description of the valve gear is there given.* Mr. Dougherty's statements have been verbally corroborated, however, by engineers who still recollect Mr. James' practice,—which, by the way, was never very extensive, while it ceased altogether in 1839. His engine of 1832 weighed about 3½ tons, and was supported upon four wheels, 3 feet in diameter, but not coupled. It had a pair of 8-inch outside cylinders by 12-inch stroke, the steam from one cylinder being, however, worked through the other. The steam ports of each cylinder were 5 inches by ¾ inch, the exhaust ports being 5 inches by 1½ inch. Each valve had ½-inch lap on each end. For each cylinder there was a forward and a backing eccentric fixed to the axle, as in Mr. James' steam carriage of 1829, already mentioned, and as now almost universally employed in all locomotive engines. The ends of the two eccentric rods in connection with each valve were jointed, at a distance of 9 inches apart, to a curved link, which could be raised or lowered as required, upon a block jointed upon the end of the valve spindle. The link was designed solely as a means of reversing the engine with certainty when in motion; "but," Mr. Dougherty writes to the author, in May, 1858, "in setting the eccentrics we found it would cut off, and we so used it on the engine to expand from different points." The reason why the lap valve was adopted has been already given, but it may be as well to repeat it in Mr. Dougherty's own words:—"You will ask how it was that Mr. James had the lap on the valves before we tried the link? The lap of the valve is older than the link by three or four years. It was one of Mr. James' hobbies to use steam twice over in all his engines,—that is, each engine had two cylinders, both of the same size, and the exhaust steam from the first cylinder was used in the second, the first cylinder having the lap valve. By so using the steam there was a considerable saving over the single engine, and I see, by some of the papers, that they are reviving the practice." The reason for this economy, if it was established, could not of course have been due to working the same steam successively through two cylinders of equal diameter, but the lap valve and probable high pressure (one or two of Mr. James' engines exploded their boilers), may have saved fuel. In referring to this instance of the early use of the link, it should be owned that it did not, in the slightest degree, open the way for the subsequent employment of that elegant form of reversing and expansive gear. In the States the link motion is commonly and very justly regarded as an English invention; and upon its re-introduction in 1849, by Thomas Rogers, the then leading American locomotive maker, it encountered considerable opposition, and it did not attain general adoption until about 1855.

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[*] This establishment was in the city of New York, and is not to be mistaken for the present extensive works of Kemble & Parrott, near West Point, sixty miles from New York.

[*] The munificent founder of the Cooper Institute, New York.

[†] Letter from David Matthew.—See ante.

In 1830, Col. Stephen H. Long, of the Staff of United States Topographical Engineers, interested himself in locomotive improvement. The result of his labours was the establishment of the Norris Locomotive Works, at Philadelphia, at one time ranking among the largest of the kind ever established. These works commenced business early in 1831, under the ownership of the American Steam Carriage Company. Their first locomotive, tried July 4, 1832, on the Newcastle and Frenchtown Railroad, was not successful. The interest of the company was afterwards purchased by William Norris, whose connection with locomotive improvement will be presently mentioned. Col. Long secured a patent, January 17, 1833, for an arrangement whereby, when two horizontal inside cylinders were employed, one slide valve box should serve for both. In the same patent, in apparent ignorance of what James had done, Long claimed the use of four fixed cams, each having nearly the action of the common eccentric. In May of the same year Col. Long patented the telescopic or "slip" chimney, for many years employed on a certain class of American engines, and subsequently applied to vessels of war by Mr. Taplin, of Woolwich. In the same patent were included means for cutting off steam at five-eighths of the stroke of the piston, the substance of the arrangement being additional lap on the valve. In December, 1833, Col. Long patented the use of a supplementary expansion valve, to cut off at half-stroke.

Historically, the engines designed by Horatio Allen, and placed upon the South Carolina Railroad early in 1832, deserve notice here. Mr. Allen (now of the Novelty Works, New York) was the first to produce an engine with a long wheel base, at the same time adapted to sharp curves. His engines, of which but four were made, consisted each of a boiler supported upon two bogies or trucks, each bogie having a pair of driving wheels and a pair of smaller carrying wheels. Each driving axle had, however, but a single cylinder working in connection with it. The driving wheels were near the centre of the engine, the smaller wheels being at the ends. The essential feature of this arrangement has been occasionally revived by various engineers, and, among these, M. Meyer, of Vienna, exhibited a design, in the International Exhibition of 1862, for a large engine which, although improved in detail upon Mr. Allen's plan of 1832, nevertheless followed the same general ideas.

The locomotive with which the Baltimore and Ohio Railroad was worked in 1831 has been already noticed. It was found too light; and the same maker shortly afterwards, or in 1832, produced a second engine—the "Atlantic"—of nearly twice the weight. This was for some years the type of the locomotives upon the line just named. The "Atlantic," made by Davis, had Cooper's vertical boiler, the dimensions of which were as follow:—

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* In the *American Railroad Journal* for 1832 (vol. i., p. 673) it is stated that James' engine was driven backwards and forwards across the yard of the maker's workshop, in Eldridge Street, a distance of 50 feet, eight times in 63 seconds, including stops. None but the most efficient reversing gear, such as the link motion is, would have secured such a result.

4 feet 3 inches in external diameter, and 69 inches high above fire grate. In the bottom was a circular fire box 46½ inches diameter and 22 inches high. From the crown plate of the fire box rose 282 iron tubes 16 inches long, and having an external diameter of 1¾ inch at their lower ends, and 1½ inch above. The smoke box was of the same diameter as the fire box, and only 6 inches deep, its upper plate being 3 inches under water, as in the boilers of some of the steam fire engines tried at the Crystal Palace in 1863. The products of combustion, after filling the smoke box, escaped through a vertical pipe 13 inches in diameter and 22 inches high, the upper portion of this pipe (19 inches) passing through the steam space of the boiler, and communicating with a 13-inch chimney, rising 14½ feet from the rails. The engine had two cylinders, placed vertically on the sides of the boiler. Each cylinder was 10 inches in diameter, the pistons having a 20-inch stroke. The connecting rods turned a shaft having a 28-inch spur wheel, which geared into another of 14 inches diameter upon the fore axle of the engine. The driving wheels were 2 feet 11 inches only in diameter, although the ratio of the gearing was intended to make them equivalent to a pair of 5 feet 10 inch wheels on the first motion. The engine was supported on four wheels of equal diameter (35 inches), although these were not coupled. The whole weight in working trim was 6¼ tons, of which 4 tons nearly were on the driving wheels. The tender, loaded, weighed 3¼ tons more. The "Atlantic" had lap valves, and the steam was worked through but two-thirds of the stroke at full pressure, the remainder of the stroke being completed by expansion. Mr. Jonathan Knight, the chief engineer of the Baltimore and Ohio Railroad, in his annual report for 1832,* estimated the friction of this engine, when moving itself only, as equal to a draught of 125 lbs.; and when drawing a load, he estimated the additional friction of the engine as 325 lbs.; making a total engine resistance of 450 lbs., or 70 lbs. per ton, exclusive of tender. The working pressure was 50 lbs. per square inch; the friction of the wagons and carriages, drawn at moderate speeds, was estimated by Mr. Knight, in 1832, as equal to 10 lbs. per ton, and he estimated the adhesion of engine wheels as one-eighth of the weight with which they were loaded. Mr. Knight found the total tractive force of the "Atlantic," inclusive of its own friction, to be 1,570 lbs. at a speed of 15 miles per hour, equal to 63 indicated horse-power. The maximum evaporation was given as 360 gallons an hour; but as these were American gallons of 8⅓ lbs. of water, this would be equal to 48 cubic feet of water. Mr. Knight preferred, however, to take the ordinary evaporation as 300 gallons, or 40 cubic feet. The waste steam of the Baltimore and Ohio engines was not discharged directly up the chimney, but against the blades of a fan, in such a manner as to cause it to revolve rapidly, and thus produce the strong blast of air requisite for the combustion of anthracite coal. This mode of working the exhaust steam must, obviously, have been attended with a considerable back pressure upon the pistons, the speed of which was from 200 to 320 feet per minute.

In the engines made subsequently to the "Atlantic" the 14-inch driven pinion was fixed upon a shaft supported in bearings upon the framing, and from cranks upon the ends of this shaft coupling rods were made to transmit the motion to one or both pairs of the wheels of the engine, as required. Another class of engines, known as the "Arabians," were of much the same construction, but were of larger dimensions. The cylinders were 12½ inches in diameter, and the stroke was 22 inches. The boiler was 4 feet 4 inches in diameter and 5 feet 4 inches high. The circular fire box was 48 inches in diameter and 22 inches high, and there were 400 iron tubes, 1¼ inches in diameter at their lower ends and 1 inch at the top, the length of the tubes being 2 feet 7 inches. The tubes passed through 2 feet of water, the remaining 7 inches of their length extending through the steam space. The fire box surface was 30½ square feet, and that of the tubes 304½ square feet. The smoke box was 4 feet in diameter and 11 inches high, and the base of the chimney was enlarged to the same diameter, the "submerged" smoke box of the "Atlantic" having been found inconvenient in respect of access to the upper ends of the tubes. The "Arabian" was first worked July 14, 1834. It weighed when in working trim 7½ tons, of which 3 tons 17 cwt. were upon one pair, and 3 tons 13 cwt. upon the other pair of wheels, both pairs being coupled. The tender, loaded, weighed 4 tons. Mr. Knight found this engine drew a load, exclusive of itself and tender, of 113 tons, upon a level, at the rate of 11·79 miles per hour; and, adopting the data taken in the case of the "Atlantic," this would be equal to 52 indicated horse-power. In 1835 an engine of the same general dimensions, but weighing 8½ tons, or, with its tender, 14 tons, drew a train weighing 113 tons up a gradient of 20 feet per mile (1 in 264) at an average speed for 5 miles of 11·48 miles per hour. Taking the data previously employed, this would require a tractive force, including engine friction, of at least 2,750 lbs., and would correspond to 84 indicated horse-power. One of this class of engines is illustrated in the first edition of Stevenson's *Civil Engineering in North America*, p. 261.

No one of the American makers already mentioned succeeded in establishing a permanent business in the construction of locomotive engines. The West Point Foundry has long since ceased to exist, although it has a namesake in full work some 60 miles from New York. Phineas Davis was killed under the wheels of one of his own engines, and his business at length passed into the hands of Ross Winans, of Baltimore. Mr. James has made no locomotives since 1839. Colonel Long assigned his interest in his inventions, at an early period, to Mr. Norris.

Of the large locomotive factories still at work in the States, the oldest is that of Matthias W. Baldwin & Co., in Philadelphia. Mr. Baldwin was the first to adopt the English locomotive practice of 1830, his earliest reproduction of which was exemplified in a working model of a locomotive made in 1831 for Peale's Museum. Placed upon a circular railway, this miniature engine drew a carriage containing five or six passengers. Its performances procured for its maker an order, in 1832, for an engine for the Philadelphia and Germantown Railroad Company. This, the "Old Ironsides," was completed in January, 1833, and weighed 5 tons. When drawing its usual train, it attained a rate of speed, for a short distance, of 62 miles an hour.[*] The "Ironsides" was a four-wheeled engine, having 9½-inch cylinders, 18-inch stroke, and 4½-feet driving wheels,—the latter being placed, as in the English engines, in front of the fire box, although soon afterwards Mr. Baldwin placed the driving wheels behind the fire box, an arrangement subsequently employed by Mr. Crampton. The cylinders of the "Ironsides" were horizontal, but outside of the smoke box; the valves were long D slides, and were worked each by a single eccentric, as in the English engines of the period. The framing was wholly outside the wheels, the eccentrics being placed on the crank axle, close to the wheels. The crank axle was nearly the same as in the English engines, except that the two cranks, corresponding to the distance from centre to centre of the cylinders, were about 3 feet 3 inches apart. The author saw the "Ironsides" at the Fitchburgh Railroad Station, in Boston, New England, as late as 1853, and a working model of it was for some time preserved at the Franklin Institute, Philadelphia. In his subsequent engines Mr. Baldwin employed a 4-wheeled swivelling truck or bogie. Fig. 48, representing one of these engines, is from the *American Railroad Journal* of 1838 (vol. vii., p. 134). They had outside cylinders, but the connecting rod worked upon a crank inside of the wheels, the framing being wholly outside of the wheels. This peculiar form of crank axle, patented by Mr. Baldwin, September 10, 1834, is shown in Fig 49. This form of crank axle, known in the States as the "half-crank," was long employed by Mr. Baldwin; and in 1848 it was extensively adopted for the 17-inch cylinder express engines, and 18-inch cylinder goods engines for the New York and Erie Railroad, the gauge of which is 6 feet. One of the latter class of engines, having six coupled 5-feet wheels and a bogie, is represented in Fig. 50, copied from a photograph taken by the author in 1851.

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* Vide *American Railroad Journal* for 1832, vol. i., p. 710

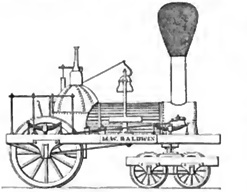

Fig. 48.—Baldwin's Engine, 1833.

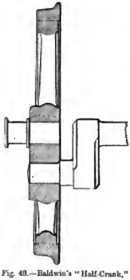

Fig. 49.—Baldwin's "Half-Crank," 1833.

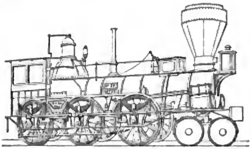

Fig. 50.—Rogers, Ketchum, & Grosvenor's Ten-wheel Engine, 1848; 18-inch cylinders, 20-inch stroke, weight 30 tons.

It may be added that other American makers who, to a considerable extent, copied Mr. Baldwin's plans, retained the same position of the crank wrist as shown in Fig. 49, but completed the crank by adding the second arm or "cheek," this arm being bedded in the cast-iron driving wheel itself. This arrangement, employed by H. R. Dunham & Co., of New York, is shown in Fig. 51,[*] and it was subsequently patented in England, July 19, 1847, by Mr. Slaughter, of Bristol.

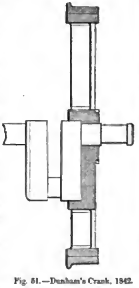

Fig. 51.—Dunham's Crank, 1842.

Mr. Baldwin's early engines had but a single eccentric for each cylinder, and the reversing gear was beneath the foot plate. The mode of reversing was substantially the same as that employed as early as 1818 by Messrs. Carmichael, of Dundee, Scotland. The cross-heads in Mr. Baldwin's engines worked each upon a large hexagonal cast-iron guide bar, which was furthermore bored out to form the barrel of the feed pump. This is shown in Fig. 12, page 16, of the present work. The pump valves were so arranged that by loosening a single screw, a strap, or "stirrup" holding the valve chambers together, was released; and all the working parts of the pump could be at once exposed to view. The most valuable of Mr. Baldwin's improvements, patented also September 10, 1834, was that of ground steam-tight joints for the steam pipes of locomotives. This was certainly no more than an obvious extension of good workmanship to parts previously made tight by red lead packing; but whoever was the first to carry out this improvement—and Mr. Baldwin appears to have been the first—deserves credit nevertheless. The engines under notice generally had 4½-feet driving wheels, and the centre of the boiler was commonly 4½ feet only above the rails. In most respects Dunham's engines, already mentioned, were like Baldwin's; and working drawings of the former will be found in the fourth volume of the *Practical Mechanic and Engineer's Magazine*. The long wheel base and low centre of gravity were subsequently insisted upon by Mr. Crampton as essential to fast engines; and express locomotives having much in common with the Baldwin engines of 1834 are still in favour on the continental railways. It may be added that in the years 1833 and 1834 Mr. Baldwin made five locomotives; in 1835, fourteen; in 1836, forty. Down to the present time 1,200 locomotives have been turned out from his shops.

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[*] Vide *American Railroad Journal* for 1833, vol. ii., p. 36. Dr. Patterson, of the University of Virginia, and Mr. Peale, were on the engine, and timed its working on this occasion. Mr. William Pettit, who ran the engine on this occasion, has informed the author that it made 1 mile in 58 seconds, and 2½ miles in 3 minutes 22 seconds.

[*] Vide *Practical Mechanic and Engineer's Magazine*, vol. iv., p. 103. *Vide* also Renwick *On the Steam Engine*. *The Practical Mechanic and Engineer's Magazine*, in connection with this subject, mentions Mr. Baldwin as having made a model of Braithwaite & Ericsson's engine in 1829, and refers to the "Old Ironsides" as having been made "soon after." The dates herein given are those which Mr. Baldwin has personally stated to the author.

William Norris, who has been already mentioned, established a locomotive engine building trade in Philadelphia at an early date. It appears from the books of the firm (Messrs. Richard Norris & Son) that one engine was made in 1832, one in 1833, one in 1834, two in 1835, eight in 1836, and so on in an increasing scale, up to one hundred in 1854. Upwards of 1,000 engines have been made in all at Messrs. Norris' works; and of these 17 were exported to the Birmingham and Gloucester Railway, and 153 to the Continent of Europe. There is no authentic description extant of the earlier Norris engines, although those made in 1836 survive in contemporary publications, and will be hereafter referred to. William Norris, the founder of the works, afterwards established a locomotive factory at Vienna; and his models were reproduced, with inconsequential alterations, by M. Meyer, who still enjoys the credit in Europe of the origination of a variety of valve gear which as will be hereafter shown, was originally introduced from the States.

It is now time to return to the progress of locomotive improvement in England. A valuable contribution to the detail of locomotive construction was made in 1835 by John Day. This was the solid wrought-iron wheel, now so generally adopted in English rolling stock. Fig. 52, from Day's specification, represents the mode of welding up a wheel from bar iron, the moulds, filling pieces, &c., being virtually the same as now employed.

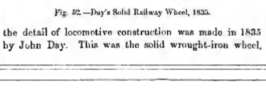

Fig. 52.—Day's Solid Railway Wheel, 1835.

This wheel has been lately improved upon by a continental engineer, M. Arbel, who stamps the whole at two heats into its finished form. It may be added here, that Day also patented, in 1835, cast-iron sleepers for railways; and he was, moreover, the inventor of the mode of cutting off the ends of railway bars by sawing them at a high heat by a rapidly revolving disc of iron. The last-named invention, first worked in 1834 at the Varteg Iron Works, South Wales, was never patented, although it is said to have diminished the price of rails 30s. per ton in one day.*

In 1836, Pambour conducted a series of experiments upon the Liverpool and Manchester engines with reference to the back pressure existing in the blast pipe. In one case he employed an arrangement whereby, while standing upon the foot plate of the engine, he could adjust the area of the blast orifice to anything between 6¼ and 2½ square inches. In the second edition of his book *On the Locomotive Engine* (1840), he recommended the adoption of this arrangement as a means of controlling the rate of combustion in the fire box. In 1836, also, Mr. Booth, of the Liverpool and Manchester Railway, patented a mode of stopping locomotive engines by closing, or partially closing, the blast pipe, the opening of which was to be regulated at will from the foot plate. The Messrs. Hawthorn employed the variable blast pipe shortly afterwards, and John Gray, then of the Liverpool and Manchester Railway, fitted it to an engine—the "Cyclops"—in 1839, the object of both Messrs. Hawthorn and Gray being to regulate the activity of the fire according to the work to be done. In 1839, also, Mr. Gray made drawings for six engines for the Hull and Selby Railway; and these, made by Messrs. Sheppard & Todd, of Leeds, had the variable blast pipe. In November, 1840, Ross Winans obtained an American patent for means for regulating the area of the blast opening from the foot plate, and in 1854 he obtained an extension of his patent for seven years. This patent was sustained in various suits at law, although the variable blast—now abandoned in all English engines, whether burning coke or coal—was originally employed in 1836.†

In 1836, also, it was thought expedient to adopt a shorter length of stroke than had previously prevailed. Ten engines were made for the Liverpool and Manchester line with 14-inch cylinders, 12-inch stroke, and 5-feet driving wheels. The capacity of the cylinders was hardly more than that corresponding to a diameter of 12 inches and a stroke of 16 inches. In all other respects the dimensions of these engines were the same, or nearly the same, as those of the ordinary class with 12-inch cylinders and 16-inch stroke. Four of the short-stroked engines were made by Messrs. Mather, Dixon, & Co.; four by Messrs. Tayleur, of Warrington; and two by Messrs. Hawthorn, of Newcastle-on-Tyne. No authentic record of the performances of these engines is now accessible, but they did not answer the expectations with which they had been constructed. Mr. Brunel, however, considered that such a proportion of stroke to diameter of driving wheel might not only be employed in the way of further experiment, but that even a relatively shorter stroke was admissible. Two of his engines for the Great Western Railway—the "Premier" and "Ariel," made by Messrs. Mather, Dixon, & Co.—had 14-inch cylinders, 14-inch stroke, and a pair of 7-feet driving wheels—a proportion of stroke to wheel of 1 to 6. These engines had 325 square feet of tube surface, and were worked at 50 lbs. pressure only. Their impuissance may be easily inferred, yet it did not prevent the extensive adoption on the Great Western line of engines of much the same proportions. Thus the "Vulcan," "Æolus," and "Bacchus," made by Messrs. Tayleur & Co., had 14-inch cylinders, 16-inch stroke, a pair of 8-feet driving wheels, and 530 square feet of tube surface. The "Neptune," "Apollo," and "Venus," made also by Messrs. Tayleur & Co., had 12-inch cylinders, 16-inch stroke, and a pair of 8-feet driving wheels. The fire-box surface was 52½ square feet, and the barrel of the boiler, 39 inches in diameter, contained 127 tubes 1⅝ inch in diameter and 8 feet 6 inches long. The "North Star," made for Mr. Brunel by Messrs. Stephenson, had 16-inch cylinders and 16-inch stroke, a pair of 7-feet driving wheels, and 640 square feet of tube surface.

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* A rather earlier application of the rail-cutting disc is attributed also to Mr. Fawell, then of the British Iron Company, Corngreaves. The whole action of the soft iron disc had been previously illustrated at the Adelaide Gallery, London. Vide *The Engineer*, vol. xvii., p. 63.

† In 1834, Mr. Bodmer patented a mode of regulating the draught in locomotive fire boxes; not, however, by throttling the exhaust steam, but by closing the base of the chimney by a sliding pipe. The effect of this was simply that of a damper; and the latter, in a different form, was often employed in the chimneys of the early locomotive engines. See MM. Flachat and Petiet's work *On the Locomotive Engine*.

In December, 1836, Mr. T. E. Harrison (now chief engineer to the North-Eastern Railway) patented an arrangement wherein the boiler was placed on a carriage separate from that carrying the cylinders and driving gear, the steam being transmitted from the boiler to the engine through a jointed pipe. This arrangement is shown in Fig. 53, which is a longitudinal section, that of the exhaust pipe being taken also (out of the general plane).

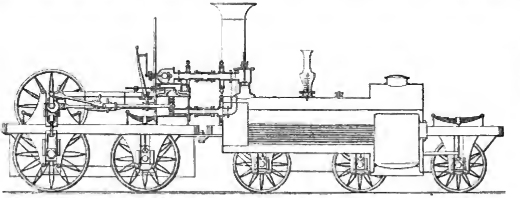

Fig. 53.—Harrison's Engine, 1837.

It will be seen that Mr. Harrison proposed to get up the speed of his engine by spur gearing, and Mr. Brunel had an engine—the "Thunderer"—constructed upon this plan, the makers being Messrs. Hawthorn. The "Thunderer" had 16-inch cylinders and 20-inch stroke, the ratio of the gearing was 3 to 1, and the driving wheels were four in number, coupled, and 6 feet in diameter corresponding to 18-feet wheels, had these been worked upon the first motion. The boiler was 44 inches in diameter, and contained 135 tubes 1⅝ inches in diameter and 8 feet 7 inches long, giving 516 square feet of surface. The fire box was of great size, the area of fire grate being 17¼ square feet, while, by the introduction of a "mid-feather" or intermediate water space, the whole fire box heating surface was 108½ square feet. The weight of this engine, empty, is given by Mr. Wood (in his *Treatise on Railroads*, third edition, 1838) as 12 tons only, although it is not stated whether this is the weight of both carriages, containing the boiler and engine respectively. It must, obviously, have been the weight of the engine alone.

Messrs. Hawthorn also, in 1837, made for Mr. Brunel another engine on Mr. Harrison's plans. This was the "Hurricane," having a boiler and cylinders of the same dimensions as those of the "Thunderer," and supported in the same manner upon separate frames, with jointed pipe connections. The "Hurricane," however, had no spur gearing, but was fitted with a single pair of 10-feet driving wheels. The chimney was 18 inches in diameter, that of the "Thunderer" being 16 inches. The "Hurricane" weighed 11 tons empty (exclusive of boiler carriage), and in addition to its 10-feet driving wheels had four carrying wheels of 4½ feet diameter each. The career of both these engines was a short one.*

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* Particulars of the performances of several of the early engines upon the Great Western and other railways may be found in Wishaw's *Railways of Great Britain and Ireland*, 1840; but it is not thought necessary to introduce them here. While describing locomotives having their boilers and engines upon separate carriages, it may be proper to refer to an engine of this class—the "Novelty"—made for the Reading Railroad, U.S., and first worked in June, 1847. The boiler, on a separate frame, had 100 tubes 2½ inches in diameter and 14 feet long. The fire box, for burning anthracite coal, extended for a considerable distance under the cylindrical barrel of the boiler, the fire grate having the large area of 36 square feet, while the total heating surface was 1,083 square feet. The fire door was in the usual position, and through this coal had to be fed upon the fire over a grate nearly 10 feet long, and extending at least 7 feet under the barrel. The fire was urged by a fan blast, and the current of flame, in turning up to enter the tubes, struck full against the dome, which was rapidly burned away. The steam was carried through jointed pipes to an engine on a separate carriage in front of the boiler. This

itself, been attributed to Stephenson, but much credit has been accorded him (and this may be freely given) for its use in the "Rocket." Considering the prior publicity of Neville's invention,* it is only explicable upon the superiority of Mr. Booth's public position that he should have been credited with the suggestion of the multitubes of the prize engine at Rainhill. Yet, after every deduction, it remains to the credit of George Stephenson that he was the first to combine the blast pipe and modern locomotive boiler in a high-speed passenger engine. There was perhaps as much in this as in the mere mental exercise involved in *scheming* a tubular boiler. For the water tubes (in a patent for which Stephenson had acquired a share in 1821) had choked with deposit; and it was impossible to pronounce, without trial, whether the fire tubes would not leak, or collapse, or choke with ashes, or burn through, or utterly stop the draught, or break apart. No amount of *à priori* reasoning upon such an arrangement would have availed, in the absence of the *experimentum crucis* of actual trial. When this took place the chief difficulty was that of leakage, and means were shortly found whereby this was effectually arrested. Had Stephenson merely constructed this boiler under the directions and at the risk of others, its success would have brought him no fame; but he staked his prospects, not only as an engine maker, but as a railway engineer, upon a mere suggestion, the value of which was to be ascertained only by himself. He assumed an immeasurably greater risk than a mere inventor could have done, and he had to consider and determine many novel details of workmanship, with which nine out of ten of the inventive fraternity would have never concerned themselves. It was thus that George Stephenson earned lasting credit in fitting fire tubes in the boiler of the "Rocket."

As a true inventor, no name stands in so close connection with the Locomotive Engine as that of Richard Trevithick. It was he who first broke through the trammels of Watt's system of condensation and low, if not negative, pressure; it was he who first employed the internal fire place and internal heating surface; he was the first to create or promote a chimney draught by means of exhaust steam—the first to employ a horizontal cylinder and cranked axle, and to propose two such cylinders with the cranks at right angles to each other—the first to surround the cylinder with hot air—the first to draw a load, by the adhesion of a smooth wheel upon a smooth iron bar—and the first to make and work a railway locomotive engine. Trevithick and George Stephenson were contemporaries;* the first locomotive seen by the latter was constructed by the former, and a personal acquaintance was afterwards established between them. Although irrelevant to the present purpose, it may be added that Trevithick patented the screw propeller, and specified several forms of that instrument, and various modes of applying it, in 1815—years before those to whom the invention is more commonly ascribed had turned their attention to it. Justice exacts the truth, however, that Trevithick's genius, brilliant as it certainly was, was of an impracticable kind, and scarcely capable of conferring any direct benefit upon society.

The next most deserving name in connection with locomotive improvement is that of Timothy Hackworth. If he discovered no important principles, he stamped a character upon the structure of the Locomotive Engine which it still retains. What he did in this respect should be ever acknowledged. It does not appear, however, that Hackworth was ever placed in a position where he had to struggle against and overcome the once strong prejudices of the public against locomotive conveyance upon railways. It is as the champion in that great contest that the name of George Stephenson must ever shine above all others; and even Trevithick and Hackworth might have felt pride in having provided, directly or otherwise, the most important aids in the final achievement of the great victory of 1829.

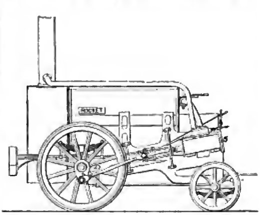

Fig. 34.—Stephenson's "Rocket," as altered subsequently to 1830, and now preserved in the Patent Museum, South Kensington.

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made April 25, 1848, by William Gowland, then an engine-driver on the London and North-Western Railway, but who, in 1829, drove the "Sanspareil" at Rainhill. Gowland states that Mr. George Stephenson asked Mr. Hackworth, in his hearing, how it was that the "Sanspareil" made such an abundance of steam, and that Hackworth described the contraction and position of his blast pipe, the orifice of which was about 1¼ inch in diameter. This was shortly after the official trial of the "Rocket," but her exhaust pipes were immediately afterwards thrown into a single pipe with a contracted nozzle.

* M. Gengembre had patented a primitive form of Neville's boiler, with four ascending and one descending tube, as early as March 6, 1821, nearly seven years earlier than Seguin. Vide *Brevets D' Invention*, vol. xxi., p. 177, plate 26.

* Trevithick was born April 13, 1771, and died April 22, 1833. George Stephenson was born June 9, 1781, and died August 12, 1848.

CHAPTER II.

HISTORY OF THE LOCOMOTIVE ENGINE—*CONTINUED*.

The sum of locomotive improvement from the date of the Rainhill competition down to the present time has undoubtedly been very great, yet it is divisible into a vast number of items, due to a most numerous body of engineers. And however great may be the contrast between the locomotive of 1829 and that of 1864, no one of these superadded improvements would by itself have imparted a fundamentally new character to the famous original. No one of them has produced a result so conspicuous as the great speed of the "Novelty," or the large evaporative power and capacity for traction of the "Sanspareil," or the happy combination of speed, power, and sound workmanship in the "Rocket." These engines, however, in consequence of restrictions imposed by a prudent regard for the strength of the light rails of the Liverpool and Manchester line, had been planned on so small a scale that they served hardly more than an experimental purpose, and the real work of the railway was, as we have seen, commenced in 1830, with engines of much the same construction as the "Rocket," but of nearly twice its weight and power. These outside-cylinder engines, with a wheel base (or distance between the centres of extreme wheels) no greater than 5 feet, ran unsteadily at what were then reckoned high speeds,—and it is to be remembered that the "Rocket" itself has made its 60 miles an hour upon an emergency. The inside-cylinder engines of the "Planet" class were therefore almost immediately adopted; and although a general return is now being made to the outside-cylinder construction, it has for a long time been customary to regard the "Planet" as the true prototype of the English locomotive engine. Although the performances of this engine have not been specifically recorded, Pambour made many experiments, in 1834, with engines of almost identical construction and dimensions.

Before proceeding to an examination of the results of Pambour's experiments, it may be as well to revert to the considerations by which all locomotive improvement is to be estimated. It need hardly be said that the "Planet" was complete as a locomotive, having its furnace, boiler, chimney, blast pipe, and a pair of high-pressure steam engines and feed pumps, all mounted upon springs upon the same frame,—the production of steam and its admission to the cylinders being under complete control. So far, too, as safety is concerned, the construction of the engines of the "Planet" class was such that they were practically as safe, at their proper working speed, as any locomotives made at the present day. The mere enlargement of an engine, by expanding all its dimensions, can hardly be reckoned an improvement in principle, although there are certain advantages, in respect of overcoming the working resistances of machines, which are best obtained on the large scale. If we take the weight of the "Planet" as 8 tons, the increase of all its dimensions by one half would, with no other alteration whatever, increase its weight as the cube of the increased dimensions, or to 27 tons; yet this increase would not conform to a proper idea of a mechanical improvement. Indeed, while the weight would increase as the cube of the dimensions, the heating surfaces and other elements of power would be increased only as the square. This point should not be lost sight of, as it is by no means uncommon to hear the increased weight of locomotives spoken of as in itself a notable gain upon the earlier practice. The best test of improvement would, of course, be the attainment of increased results without increased weight.

The improvements made since 1830 in locomotive engines may be justly regarded as, *first*, the production of increased mechanical power in proportion to the weight of the machine; *second*, greater durability in proportion to the work performed; *third*, greater economy of fuel. Increased mechanical power may be directed to the overcoming of a given resistance at an increased speed, or to the overcoming of a greater resistance at a given speed; although much better arrangement and structure are requisite for great speeds than for great traction. The work performed by a locomotive engine can be properly expressed only by the product obtained by multiplying the total resistance into the distance through which it is overcome in a given time. For the smallest engine, if its wheels are prevented from slipping, may draw a very great load, if speed be left out of consideration. In other words, only the same mechanical power is required to overcome a resistance of say 1,000 lbs. at the rate of 1 foot per minute, as would be required to overcome that of 1 lb. at the rate of 1,000 feet per minute. It is usual to express all resistances overcome by mechanical power in avoirdupois pounds, and to express the rate at which they are overcome in English feet per minute. The product of these measures the mechanical power in "foot-pounds;" and the latter may be divided by 33,000, with a quotient expressed in "horse-power," as a more convenient standard of comparison wherever a great number of figures would be otherwise required. The term "nominal horse-power," which bears no assignable proportion to the real horse-power of fixed and marine engines, has not, happily, been adopted by locomotive engineers. But the actual power of a locomotive engine can only be properly expressed in either foot-pounds or horse-power, and yet these terms, indispensable to a just comparison of locomotive efficiency, are as yet but little employed for the purpose.

Pambour made the performances of the early locomotives the subject of much original and useful analysis; but he did not ascertain their actually exerted power in either foot-pounds or horse-power, nor did he recognize the importance of these standards. He has, nevertheless, given data from which the horse-power exerted in a considerable number of trials may be approximately estimated—although only approximately, upon the supposition that the speed was nearly uniform over the whole length of the line. This supposition, although manifestly inadmissible as correct, is not probably so far from the truth as seriously to affect the results about to be deduced. Pambour's data are scattered throughout his book, but they may be collected as follows:—

DIMENSIONS AND PERFORMANCES OF LIVERPOOL AND MANCHESTER LOCOMOTIVES, 1834.

Name of Engine.	Diameter of Cylinder.	Length of Stroke.	Diameter of Driving Wheels.	Weight of Engine, Loaded.	Weight of Train, including Tender.	Average Speed in Miles per Hour over whole Line.	Probable Resistance per Ton of Engine and Train.	Total Resistance.	Equivalent Work done in Horse-power.
	Inches.	Inches.	Feet.	Tons.	Tons.		Lbs.	Lbs.	
Vulcan,	11	16	5	8·34	39·07	23·92	19	901	57·3
Atlas,*	12	16	5	11·40	195·50	9·72	10	2,069	53·6
Do.,	12	16	5	11·40	127·64	15·00	12	1,668	66·7
Do.,	12	16	5	11·40	40·15	15·53	15	773	32·0
Fury,	11	16	5	8·20	56·16	19·07	16	1,030	54·0
Do.,	11	16	5	8·20	48·80	18·63	16	912	45·0
Firefly,	11	18	5	8·74	41·40	18·63	16	802	39·6
Do.,	11	18	5	8·74	41·40	22·70	18	903	54·7
Vesta,	11½	16	5	8·71	33·15	27·23	21	879	63·8
Leeds,	11	16	5	7·07	88·34	18·63	15	1,431	70·6
Do.,	11	16	5	7·07	37·51	22·84	18	802	48·8

* Six-wheeled Engine; four wheels coupled.

The speeds are those while the engines were in motion, and are exclusive of time lost in stoppages. The estimated resistances per ton of engine and train are assumed for the present purpose, and are much beyond those deducible from Pambour's own conclusions, which pointed rather to one and the same resistance of trains at all speeds upon a level, viz., 8 lbs. per ton. What the resistances per ton really were in the above trials it is, of course, impossible to say; but those assumed are believed to represent fairly what would be due to the friction of the engines, carriages, and wagons, the jolts due to the irregularities of the line, and the slight average effect of gravity and curvature upon the Liverpool and Manchester Railway, which is nearly level. The foregoing table will derive further interest from the following, which includes additional particulars relating to the same trials:—

EVAPORATIVE POWER OF LIVERPOOL AND MANCHESTER LOCOMOTIVES, 1834.

Name of Engine.	Area of Fire Grate.	Area of Fire Box Surface.	Area of Tubes.	Horse-power actually exerted, as already estimated.	Water Evaporated per Hour.	Best Worsley Coke burnt per Hour.	Water Evaporated per Lb. of Coke.	Average Pressure of Steam per Square Inch.
	Square Feet.	Square Feet.	Square Feet.		Cubic Feet.	Lbs.	Lbs.	Lbs.
Vulcan,	6·50	34·45	307·38	57·3	59·95	538½	7·00	54·5
Atlas,	9·20	57·06	217·88	53·6	43·56	526	5·18	52·7
Do.,	"	"	"	66·7	48·30	622	4·85	53
Do.,	"	"	"	32·0	46·52	463½	8·27	30*
Fury,	6·12	32·87	307·38	54·0	52·03	537	8·05	57
Do.,	"	"	"	45·0	55·03	471	7·30	57
Firefly,	7·16	43·91	362·60	39·6	62·18	555	6·99	44
Do.,	"	"	"	54·7	74·34	669	6·99	49
Vesta,	7·06	46·00	256·08	63·8	61·00	709	5·34	61
Leeds,	6·19	34·57	307·38	70·6	60·52	586	6·68	54
Do.,	"	"	"	48·8	65·87	534	7·70	49

* Pressure purposely kept down in this experiment.

We have divided the consumption of fuel and water over the actual times during which the engines were in motion, instead of over the whole period occupied on each trip inclusive of stoppages. In most cases, however, the engines ran the whole distance without stopping. Pambour estimated that, of all the steam generated in these engines, one-fourth was wasted by being blown away at the safety valves—a loss which would, of course, have been attended by a corresponding loss of fuel. It is not probable, however, that this loss bore a constant proportion to the whole evaporation in each case. If there were reasons for believing that it amounted to one-fourth of the whole evaporation, on the average, there are equal reasons for believing that it amounted to much more in some cases and to much less in others. This view, which derives much support from probability, will assist in explaining the apparent discrepancies in the above tables. And as the whole question of the performances of these engines turns upon the actual mechanical power exerted by them, it is important, in the absence of positive knowledge of the actual resistances, to exercise care in estimating them from the general conditions of the experiments. It is preferable, indeed, to assign an even higher value to these resistances (and thus to give the engines credit for greater efficiency than would be reckoned probable from later experience) than to under-estimate their performances. It is with this conviction that the resistances have been estimated at their probable *maxima*. The consumption of fuel and water in some of the trials would appear, however, to justify an even higher estimate, and it will be well, therefore, to examine the trials more in detail.

Taking the first engine in the table, the "Vulcan," and adopting Pambour's estimate of one-fourth as the loss of steam blown away, the useful evaporation was 59·95 × ·75 = 44·96 cubic feet of water per hour. A pair of 11-inch cylinders with 16-inch stroke would take 3·5197 cubic feet of steam at each revolution of the driving wheels, irrespective of the loss of steam by the clearance at the ends of the cylinders and in the steam ports. We may therefore reasonably take 4 cubic feet of steam as the real expenditure, inclusive of these losses, at each revolution of the driving wheels. For it is to be remembered that the valves were practically without lap, and that steam was admitted during the whole stroke. A 5-feet wheel makes 336 revolutions in running a mile, irrespective of slipping, if any; and in running 23·92 miles an hour, 8,037 revolutions would be made in the same time. We have, then, an expenditure of 32,148 cubic feet of steam per hour in the case of the "Vulcan." Now, to evaporate 44·96 cubic feet of water (the quantity usefully evaporated after allowing for the loss by blowing off at the safety valves) into this bulk of steam, the corresponding pressure could not exceed 23 lbs. per square inch above the atmosphere. The pressure in the boiler was 54·5 lbs.; but there is no fixed relation between the pressure in the boiler and that in the cylinders. The deduced pressure of 23 lbs. in the cylinders is obtained by dividing the bulk of steam expended by the bulk of water evaporated in producing it, or 32,148 ÷ 44·96 = 715; and, by referring to a table of the relative volumes of steam and water at different pressures, we find 715 to correspond to steam of a total pressure of about 38 lbs., or 23 lbs. above the atmosphere. But steam of 23 lbs. pressure, even if we make no allowance for back pressure, would, in the particular case of the "Vulcan," give a tractive force of only 742 lbs., whereas we have assumed the resistances as 910 lbs., and the total force exerted must, if the engine move at all, be necessarily equal to the resistances overcome. It follows, therefore, either that the resistances have been reckoned as even greater than they really were, or that Pambour's estimate of a loss of one-fourth of the steam at the safety valves was too high. It is evident, in either case, that the "Vulcan" has been credited in the above tables with all the work she accomplished, viz., 57·3 horse-power.

We will next take the "Atlas." Her 12-inch cylinders with 16-inch stroke took 4·19 cubic feet of steam at each revolution,—or say 4½ cubic feet, to allow for clearances and losses in ports. The driving wheels, at 9·72 miles an hour, made 3,266 revolutions in that time, and the expenditure of steam, therefore, at the average pressure maintained in the cylinders, was 14,697 cubic feet. Allowing one-fourth of the steam generated to have been blown away, the useful evaporation in the first experiment was 32·67 cubic feet, the relative volume of the steam, as worked in the cylinders, being therefore 450, corresponding to a pressure of about 48 lbs. per square inch above the atmosphere. But steam of this pressure, even without making any allowance for back pressure against the pistons, would give the "Atlas" a tractive force of but 1,843 lbs., whereas we have estimated the total resistances overcome, exclusive of back pressure against the pistons, as 2,069 lbs. Here, again, the engine has been credited with more work than was really performed, or, otherwise, the loss of steam by blowing off was less than Pambour estimated.

In the second trial of the "Atlas" the density of the steam in the cylinders, calculated in the same manner, would correspond to but 29 lbs. per square inch, which, without allowance for back pressure, would give a tractive force of only 1,114 lbs., whereas 1,668 lbs. have been allowed for the resistances overcome, although in no case could they have been either more or less than the tractive force, as referred to the pistons.

The third trial of the "Atlas" presents a striking instance of wasted steam. Although the load was of but about one-fifth the weight of that taken in the first trial of the same engine, and although the speed was hardly more than one-half greater, the evaporation of water was considerably more. The boiler pressure in the third trial had been purposely kept down to 30 lbs., and from this very fact we may judge the blowing off to have been very great. Indeed, with the exception of the time occupied (about 12 minutes) in ascending an incline of 1 in 89, and 1¾ miles long, the average pressure was but 25 lbs. The effective pressure upon the pistons could have been known only with the aid of an indicator, and this was not employed by Pambour. The valves, however, did not allow the steam to escape from the cylinders until the full stroke of the pistons had been completed, and the blast pipe of the "Atlas" was $2\frac{11}{16}$ inches in diameter; so that, with 25 lbs. steam in the boiler, the mean pressure upon the pistons, after allowing for the back pressure, could hardly have exceeded 20 lbs. This gives a tractive force of 768 lbs., although upon the short gradient of 1 in 89 an additional force of 1,297 lbs. would be required to overcome the gravity. A part of this was, however, supplied by the momentum of the train going at full speed on striking the gradient. Our estimate, therefore, of 773 lbs. as the whole mean resistance for the entire trip cannot be far wrong.

The performances of the "Firefly" would appear to be considerably less than they should have been for the fuel and water expended. The engine was, however, in bad order, and was about to be repaired. Thus the internal resistances of the engine were probably greater than we have estimated. Like most of the other engines, it was helped over the only considerable gradient in either direction of the line. But it may be better to leave its trips out of the general comparison. The second trip of the "Leeds" also shows inferior results. The fact that the apparent evaporation was 6·68 lbs. of water per pound of coke on one trip, and 7·7 lbs. on the next, goes to show, what Pambour states was the fact, that this engine was liable to priming. To say nothing, therefore, of the loss of steam by blowing off (and the trip was made with the regulator only partly open), it is evident that a considerable portion of the water apparently evaporated passed, along with the steam, through the cylinders. Very likely the useful evaporation did not exceed 50 cubic feet per hour; indeed, this would hardly allow for Pambour's estimated loss of one-fourth by blowing off. But at 50 cubic feet the volume of steam worked through the cylinders would be 614 times greater than this, corresponding to a pressure of about 30 lbs. per square inch. From this the back pressure, especially while the engine was priming, would be considerable, so much so that probably 25 lbs. would represent the average effective pressure. The blast pipe, we may add, was smaller than that of any other engine tried—viz., 2⅛ inches. An effective pressure of 25 lbs. would give the "Leeds" a tractive force of 807 lbs., whereas our estimate of the actual resistance is 802 lbs.

On the whole, therefore, it may be admitted that the power has not been under-estimated; and as this power includes that spent in overcoming the internal as well as external resistances of the engine itself, exclusive only of back pressure against the pistons, we have the probable indicated horse-power of the engines upon which Pambour experimented. Leaving out the "Firefly," as an engine out of repair, we have the following results:—

Name of Engine.	Best Coke consumed per Indicated Horse-power per Hour.	Coke per Mile.	Coke per Ton of Engine, Tender, and Train, moved One Mile.
	Lbs.	Lbs.	Lbs.
Vulcan,	9·4	22·5 Pass.	0·42
Atlas,	9·8	54·1 Goods	0·26
Do.,	9·3	41·5 ,,	0·30
Do.,	14·5	30·0 ,,	0·58
Fury,	9·9	27·3	0·42
Do.,	10·5	25·3	0·44
Vesta,	11·1	26·0	0·62
Leeds,	8·0	30·4	0·32
Do.,	10·9	23·4	0·52

The "Atlas" was a goods engine; the others were made for passenger traffic; but in Pambour's trials several of them were attached to light goods trains.

The best Liverpool and Manchester engines, therefore, of the year 1834, consumed at least 10 lbs. of the best coke per indicated horse-power per hour, and perhaps 11 lbs. or 12 lbs. would be nearer the actual consumption. For a large part of this consumption, however, corresponding to the steam blown off at the safety valves, the engines were not in fault. Any engine, however good, might be made, by extravagant firing, to waste an equal proportion of the fuel. It is, indeed, difficult to credit Pambour's estimate of the actual extent of this loss; but, for the credit of locomotive engineering in his time, we will not refuse to admit his conclusion upon this point. We shall then have a net consumption of say 8 lbs. of coke per indicated hourly horse-power, and we shall see further on how this compares with later engines. In but one case, too, was 10 indicated horse-power obtained for each ton of the weight of the engine. The general average was about 7 horse-power per ton. The average with modern engines is perhaps 12 horse-power per ton, and it often rises to 20 and even 25 horse-power per ton. In only one of Pambour's experiments of 1834 was the total evaporation as much as 2 cubic feet per hour for each square foot of fire box surface, and in but one case was the evaporation as much as one-fifth of a cubic foot per hour per square foot of the total heating surface in the fire box and tubes.

The cost of repairing the Liverpool and Manchester engines appears, in comparison with the prevalent expenditure under the same head now, to have been excessive. It is true that the repairs of a small stock of engines were then conducted under many disadvantages, and that both the labour and materials employed in repairs were inferior in quality and dearer in price than those to which our locomotive superintendents are now accustomed. At the same time, however, the original cost of an engine and tender of the period was but from £700 to £800; the "Leeds" and "Firefly," which figure in the foregoing tables, having cost but £790 each. The line was opened in September, 1830, and therefore, even in 1834, most of the engines must have been comparatively new. They were small, simple in construction, and were worked at very moderate speeds. Yet in the half-year ending June, 1832, the cost of locomotive repairs was 9·3*d.* per train mile; in the next half-year it was 10·8*d.*; in the next, 10⅜*d.*; in the next, 11·17*d.*; and for the half-year ending June, 1834, the cost of repairs was 13·1*d.* per train mile. There were in 1834 thirty locomotives on the line, but of these, Pambour says, ten were useless. Yet the sum paid for repairs in 1833 would have purchased twenty new engines at £778 each, which was then a fair price. The repairs for the first half-year of 1834 cost equal to twelve new engines. Pambour estimated the cost of repairs, on the average, as 0·342*d.* per gross ton moved 1 mile on a level, equal to at least twelve times the present average cost. The mode of making these repairs explains this excessive cost, while it also shows how defective must have been the engines when new. Pambour says (writing in 1834)—"When an engine requires any repair, unless it be for some trifling accident, it is taken to pieces, and a new one is constructed, which receives the same name as the first, and in the construction of which are made to serve all such parts of the old engine as are still capable of being used with advantage. The consequence of this is, that a re-constructed or repaired engine is literally a new one. The repairs amount thus to considerable sums, but they include also the renewal of the engines." Such remarks, applied to engines but two or three years from the makers' workshops, proclaim such defects of original structure as to make it almost a matter of wonder that the Liverpool and Manchester line was worked at all. The cost of fuel was seldom one-half, and in 1834 was not one-third that of engine repairs; nor were the repairs of the permanent way, of which so much has been said, so costly, at any time down to the middle of 1834, as locomotive repairs, which amounted annually to a sum equal to about 2 per cent. upon the total share capital of the undertaking.

It has been the custom with many authors to speak of the consumption of fuel in locomotive engines as if it formed the only consideration in determining the question of working excellence. But while the early

Liverpool and Manchester engines burned between two and three times as much fuel per horse-power per hour as suffices now, their repairs, referred to the same standard, were at least twelve times more costly than those now made upon engines of the present construction. The improvement made during the last thirty years in the structure and workmanship of locomotives has been productive, therefore, of a much greater economy of working than has attended that other class of improvements, of greater interest to the student and mechanical philosopher,—to wit, improvements in the generation, distribution, and working of the steam in locomotives.

The defects of the engines of 1830-34 were many. The tubes were at first made of copper, and these were rapidly cut through by sharp particles of coke carried through them under the force of a powerful steam blast. Mr. John Dixon, now the chief engineer of the Stockton and Darlington Railway, was the first, in 1833, to recommend the employment of hard brass instead of copper for tubes; and this single change has effected an incalculable saving in the repairs of coke-burning engines. The wheels of the early engines were made with wooden spokes fixed in cast-iron naves. The tyres were not equal in quality to those now made. The frames were rudely constructed of timber and iron, fastened together by rough bolts. The wearing surfaces were small in proportion to the strains they were made to withstand, and the refinements of steel bushings and case-hardened surfaces were unknown. The wheels were without counterweights, and the tyres were coned to an extent which, while it caused constant and most trying strains to the whole machine, disclosed great ignorance of the actual movements of locomotives upon straight or slightly curved lines, especially with a wheel-base only 5 feet in length, and with widely spread cylinders and unbalanced disturbing weights. The eccentrics were loose on the crank axle, and were held to their work only by stops, or "drivers," as they were called, and the shock in reversing, especially at high speeds, was great. It was common, too, to attach the draw-iron directly to the fire box, and thus to discharge upon the boiler all the shocks due to inequalities in the draught of the train. The fire bars were widely spaced, and did not fit accurately to the sides of the fire box, and the consequence was that much fuel was lost,—the more so as there was neither ash pan nor damper. In many of the early engines the cylinders were fastened chiefly to the smoke box, and the boiler was so fastened to the frame that, when heated, the former could only expand under considerable strain. The chimneys were too large, generally 13½ inches in diameter for engines with 11-inch cylinders. The blast pipes were small, generally 2½ inches in diameter, and they were carried to some distance up the chimney. In all cases the steam followed for almost the entire length of the stroke of the pistons; and, as the valves had no lead on the exhaust, the back pressure, as well as loss of expansive effect, was considerable. The engines, too, received more severe usage at the hands of their attendants than would be tolerated now. They regulated the firing and feed-water with less care than is reckoned indispensable now; and if they did not burn the fire boxes they often caused the tubes to leak. It was common, when the supply of steam permitted, to run with the fire door open, and the engines were often reversed with a full head of steam on the pistons. The piston packing did not always receive proper attention, and the matter of lubrication was not as carefully regarded then as now. The boilers, too, were often blown off with a fire burning in the fire box; and in many ways the cost of repairs was swelled by improper management. The engines worked, too, at a great disadvantage by reason of the low pressure, 50 lbs. per square inch, which had been adopted from considerations of safety. The boilers, small as they were, and of iron of good thickness, ought to have been safe at 100 lbs., and the working parts of the engine should have been equal to the increased strain. At the higher pressure the boilers could have evaporated rather more water in a given time, while, on the other hand, at the low pressure employed, at least half of the steam was lost in atmospheric resistance and back pressure upon the pistons. So, too, the working pressure was too low for useful expansive working, even had it been attempted.

Such an enumeration of the defects of the locomotives of 1834 will serve to indicate the general direction of subsequent improvement. It was soon found that engines on four wheels only were objectionable because of the considerable weight of the driving wheels upon the rails, and because of the unsteadiness with a short wheel-base, as well as on account of the increased danger to the engine and train in case of the breaking of an axle. The adoption of six wheels was no more than a return to Hackworth's practice of 1827, if we may not, indeed, refer to the eight-wheeled Wylam engines made many years earlier. Hedley's and Hackworth's object in employing six and eight coupled wheels was, of course, no more than a better distribution of weight; whereas in the Liverpool and Manchester engines extra bearing wheels were still further desirable on account of increased steadiness and safety at considerable speeds. Indeed, on the Liverpool line, the addition of a pair of trailing wheels behind the fire box did little to relieve the rails, since only a few hundredweights were made to bear upon each of them, the load upon the driving wheels being practically as great as before. The rails originally laid down upon the Liverpool and Manchester line were of the "fish-bellied pattern," and weighed 35 lbs. per yard only, and had been adopted in 1829, when the directors were debating whether the railway should be worked by stationary engines or locomotives of only 6 tons weight. Even with engines no heavier than 9 tons it was soon found that stronger rails were required; and even, also, after an extra pair of wheels had been added, it was found requisite to put down new rails weighing 66 lbs. per yard. Some portion of the excessive cost of repairing the earlier engines is, indeed, to be attributed to the jolts they experienced in running over a light and badly shaped rail supported upon stone blocks.

In the meantime various makers were constructing locomotives having special peculiarities in the arrangement of their parts; although few, if any, embodied any essential departure from the principles exemplified in the "Rocket" or the "Planet." Mr. James Kennedy (then directing the factory of Mr. Bury, and who was afterwards associated with him in the firm of Bury, Curtis, & Kennedy, of Liverpool) had brought out, in 1830, the prototype of a class of engines which afterwards became favourably known for their compact structure and comparatively good working qualities. They had inside framing, as all the Stockton and Darlington engines had; but the framing, as well as the working parts generally, was better in design and workmanship than that of the locomotives which preceded it. Mr. Kennedy's engine, the "Liverpool," was the first with driving wheels so large as 6 feet in diameter: this was built in 1830, having been commenced in October, 1829, as appears from a paper read by Mr. Bury, in 1840, before the Institution of Civil Engineers. The later engines by the same maker had smaller driving wheels, generally 5 feet in diameter. An exact drawing of one of these is still extant in the first edition of MM. Flachat and Petiet's work on the Locomotive Engine, from which Fig. 35 is copied.

Mr. Kennedy preferred the round-backed and dome-topped fire box, as shown. Several of these (Bury's) engines were sent abroad; and to this day the same general structure and arrangement of framing is retained by the American locomotive makers. It will be observed that our illustration of Bury's (or Kennedy's) engine shows a steam whistle. It is uncertain by whom the cup whistle was first made; but it is believed to have been the invention of William Stephens, a workman at the Dowlais Iron Works, in South Wales, where it was in use in 1833. A steam whistle of the form of the sportsman's whistle was in use much earlier. Messrs. Sharp, Roberts, & Co., of Manchester, had occasion in 1835 to send one of their workmen to Dowlais to put up some shafting. This workman (Thomas Turner, now or lately employed at Messrs. Beyer, Peacock, & Co.'s, Manchester) brought home with him a section of Stephens' whistle. His foreman, Mr. Rose (now engineer to the Crystal Palace, Sydenham), showed the section to Mr. Fyfe, then locomotive superintendent at the Manchester end of the Liverpool and Manchester line. Mr. Fyfe adopted it at once, and its use soon became general on all railways. Turner's pencil sketch is still preserved by Mr. Beyer, of Manchester; and from this Fig. 36 has been reduced. Fig. 37 shows the form of the whistle for stationary boilers as used at Dowlais.

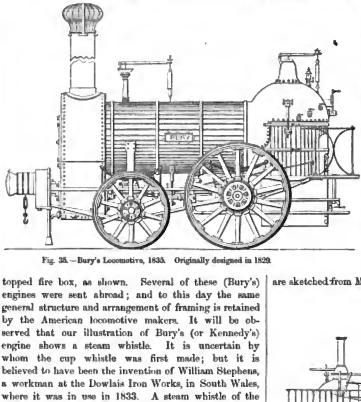
Fig. 35.—Bury's Locomotive, 1833. Originally designed in 1829.

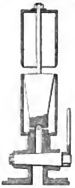
Fig. 36.—First Locomotive Steam Whistle, 1835.

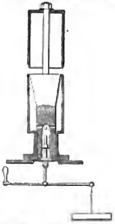
Fig. 37.—Steam Whistle at Dowlais Iron Works, 1833.

Mr. Richard Roberts, of the firm of Sharp, Roberts, & Co. (whose establishment has since become the property of a Company formed under the Limited Liability Act—to wit, Sharp, Stewart, & Co., Limited) turned his attention to locomotives as early as 1832. In 1833 he designed a class of engines of which one, the "Experiment," was worked upon the Liverpool and Manchester line, and three were sent to the Dublin and Kingston Railway. These engines are represented in Figs. 38 and 39, which are sketched from Mr. Roberts' original drawings.

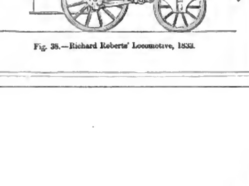
Fig. 38.—Richard Roberts' Locomotive, 1833.

Mr. Roberts' first engines had outside vertical 11-inch cylinders; the piston rod working through the top cover, and connected, by a cross-head and pair of side links, to a bell crank lever, made in pairs, from which the power was transmitted to the driving wheels. The slide valves were cylindrical, and of a kind patented by Mr. Roberts in 1832. Fig. 40 will serve to illustrate their construction. The valve, of wrought iron, was formed of two concentric tubes or pipes, *a* and *b*, the larger pipe, *b*, having holes perforated at *o*, to admit steam from the steam pipe into the annular space between *a* and *b*. This annular space was closed steam-tight at each end of the valve, and steam could only escape from it alternately to each end of the cylinder through the slots *g* and g^1. The exhaust steam passed from one end of the cylinder directly into the waste pipe, and from the other end it traversed the interior of the pipe *a* of the cylindrical valve. These valves did not work well, as they did not expand equally with their cast-iron casings when heated by steam. For this reason the cylinder valves were soon abandoned. It should be mentioned that in Mr. Roberts' first engines, the valve for each cylinder was worked with a motion derived from the opposite side of the engine. No eccentrics were employed, the requisite motion being taken from a pin near the fulcrum of each bell crank (see Fig. 39), and transmitted thence, through suitable gearing, to the valve attached to the cylinder on the opposite side of the engine.

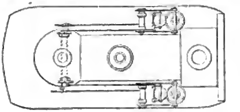

Fig. 39.—Roberts' Locomotive. Plan.

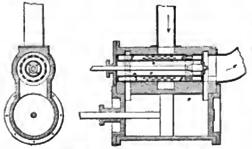

Fig. 40.—Roberts' Cylindrical Valves and Solid Pistons, 1832.

In the same specification in which Mr. Roberts described his cylindrical valves he described also a system of variable expansion gear which, although it was never applied by himself to locomotives, was afterwards extensively adopted by many builders, especially in America. A supplementary expansion valve worked in a casing at the back of the principal steam valve. This supplementary valve was worked by an eccentric, or other means, and through the intervention of a curved arm, upon which arm a sliding block, jointed to the valve spindle, could be adjusted at will, either near to or far from the axis of the motion, so as to give a shorter or longer range of motion to the expansion valve, according as it was desired to cut off the steam earlier or later during the stroke of the piston. Mr. Roberts' drawing, attached to his specification, shows the application of this system to fixed engines, and in these the block on the curved arm employed to work the expansion valve was to be connected with the governor of the engine, so as to vary the expansion according to the work being done at any moment. In locomotives the adjustment was to be effected by hand. Fig. 41 is from the specification,—a multitude of working joints being shown, although these in no way affect the principle upon which the action of the expansion valve depends. A is the steam cylinder; E, the cylindrical steam valve; I, the cylindrical expansion valve; H, the eccentric rod for the main steam valve; J, eccentric rod for expansion valve; K, arm worked by J; Q, curved arm working with K on fulcrum L. A sliding block at the end of the connection, M, may be adjusted on the curved arm, Q, either by the governor as shown, or by a handle with notches, shown by dots. The rod M works the expansion valve.

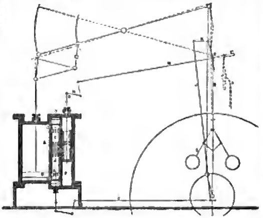

Fig. 41.—Roberts' Variable Expansive Gear, 1832.

This description of Mr. Roberts' expansion gear would be premature here, did not chronological justice require it. Although devised in 1832, nothing of the kind was applied to locomotives until many years afterwards, and then only in connection with other names. As has been already seen, expansion gear was a needless refinement upon locomotives worked at a pressure of but 50 lbs. per square inch, and with considerable back pressure. Yet even in 1832, as now, the work of locomotives was variable upon different portions of the line, and with trains of inconstant weight, moved under all conditions of weather. Knowing this, Mr. Roberts justly observed in his specification, that "with a given density of steam and a uniform

In whatever department of history, the authentication of facts demands only industry on the part of the historian. It requires, however, a degree of discrimination to decide what facts may be raised to the dignity of historical events. As incidents trivial in themselves may determine the destiny of a nation, so in engineering a single unobtrusive departure from current practice may instigate a revolution the results of which will endure for all time; while a conspicuous eccentricity may be soon forgotten. Mr. Harrison's engines were extraordinary in 1837, and they would be equally so now. Even a farm labourer might have wondered at the difference between them and other locomotive engines; and yet but one or two historians in this department of engineering have condescended to notice them, if, indeed, others have had any knowledge of their former existence. In 1837-38, however, three most valuable improvements were, in a few instances, introduced into the English locomotive engine; and although even a practical engineer might have overlooked any one or all of them in a casual survey,—and to this day there is some controversy as to their origin,—they have nevertheless permanently altered the habits, if not the character, of the locomotive. These improvements were the lap valve, fixed eccentrics, and counterweights in the driving wheels. Neither of these was patented, at least in England, nor was the influence of either widely recognized at the time, nor indeed for some years afterwards, although there are those who now rank them in importance with the blast pipe and multitubular boiler.

It is not known who first learned the effect of, and practically adopted, lap on the slide valve. Watt's assistant, Murdoch, employed the long slide valve in 1784; and, indeed, Hero of Alexandria describes a sliding valve in his treatise on pneumatics, written 2,000 years ago. Whoever at an early date employed the slide valve for steam engines—whether Murdoch, Murray, Freemantle, Trevithick, or Stephenson—must have known that unless its length agreed exactly with the distance between the outer edges of the steam ports, there would be a space in the length of each stroke of the piston during which no steam could enter the cylinder. The very knowledge required in making the slide valve as Murray made it involves this knowledge also. Watt, and almost every engineer after him, understood too the theory of expansion; and expansion in the steam engine requires the very exclusion of steam which the lap valve, properly adjusted to the motion of the piston, necessarily causes. It is, nevertheless, the fact that Tredgold, in the first edition (1828) of his work *On the Steam Engine*, regarded the slide valve moved by an eccentric as incompatible with expansive working. Tredgold described means by which the slide valve might receive two movements at each stroke of the piston, so as to cover both steam ports during a certain interval, and afterwards to open one or the other, as required, at the beginning of the succeeding stroke. This motion was for many years afterwards employed in America in locomotives made by Ross Winans, who patented a kind of valve gear, July 29, 1837, giving to the slide valve the movement proposed by Tredgold.* In America, too, as has been already mentioned, James, Davis, and Long applied lap on each end of the slide valve with a view to expansion, in 1829, 1831, and 1833 respectively. There was a difficulty, however, in starting a heavy train with an engine having lap on the slides; and the lap valve was but little employed in America until after its re-introduction from England, about 1840. The early Liverpool and Manchester engines had from $\frac{1}{16}$th to $\frac{1}{8}$th-inch lap at each end of their valves, merely to avoid the chance of the admission of steam to both ends of the

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engine consisted of a pair of 18-inch cylinders, with 20 inch stroke, driving eight coupled wheels, each 3 feet 10 inches in diameter. Upon the same frame was a close plate-iron cylinder 3½ feet in diameter and 18 feet long, which served as a condenser for the exhaust steam and for heating the feed water, while the weight thus added to the engine was serviceable for adhesion. The whole weight of the engine carriage (or exclusive of boiler carriage) was 21¾ tons. Although the steam was condensed to a considerable extent, no advantage was taken of this circumstance to obtain a vacuum. The gross load of the "Novelty," as of the other engines of the same weight upon the line, was 75 loaded coal wagons, weighing 750 tons, over a railway 95 miles long, and having a total fall of 600 feet in that distance,—the rate of descent being nowhere greater than 1 in 264, while there was no ascent in the direction of the coal traffic, except for a short distance, where an assistant engine was employed. The usual speed was about 10 miles an hour, and the consumption of coal 126 lbs. per mile. The evaporation of water could not be estimated on account of the return of a large part of the condensed steam to the boiler. The "Novelty" was only worked for a short time. It may be added that it was an infringement of Mr. Harrison's American patent, dated 1837.

* The records of the early French inventions in the steam engine show that, although the circular eccentric was not always used, the French engineers were in the habit of adding lap to the short slide valve—a lap often equal to the width of the steam ports. Hamy's engines, vide *Brevets d'Invention*, vol. xli., p. 104, and plate xi., were patented in 1827, and had the full modern allowance of lap on the slides, although these were worked by a tappet motion, much as afterwards proposed by Tredgold. Hamy, in his specification, proposed to cut off at two-thirds of the stroke of the piston, thus working expansively through the remaining third of its length. A better arrangement was patented in 1829 by M. Pecqueur, whose engines are described in the *Brevets d'Invention*, vol. xxxix., p. 201, plate xxiii., and also in the *Bulletin de la Société d'Encouragement* for 1835, p. 480. M. Pecqueur employed a three-sided cam to work his slide valve, although this cam was properly described by him as an "eccentric." He gave to the valve a lap at each end equal to the width of the steam ports, and with this and the irregular motion of his eccentric cut off the steam at one-third of the stroke of the piston, leaving the steam already admitted to expand during the remaining two-thirds of the stroke. Pecqueur's engines were not, however, publicly described at the time they were patented, nor until 1835; and Hamy's specification, as in the case of all French patents, was not published until the official protection had expired, say in 1837. Galain Dusart adopted much the same arrangement as Hamy's in his engines, patented July 3, 1832; vide *Brevets d'Invention*, vol. xxxvi., p. 124, plate xvi. The lap here was also equal to the full width of the steam ports. The addition of lap to the slide valve effected one of the most valuable ameliorations to which the steam engine has ever been subjected. It is greatly to be regretted that the whole history of the lap valve cannot be given here, and that only the fragments already presented are all that have been recovered. It is probable that lap was first added without a clear knowledge of its real effect, and that, therefore, its value in expansive working was a chance discovery. Oliver Evans is said to have applied lap valves, but this is not authenticated to the author's satisfaction. One of the last works upon which Evans was engaged was an engine for the Fairmount Water Works, at Philadelphia, and started December 15, 1817. It had a single 20-inch cylinder and 5 feet stroke, supplied from four plain cylindrical boilers 30 inches in diameter and 24 feet long. The ordinary working pressure was 200 lbs. per square inch, although 220 lbs. was occasionally carried. With 194 lbs. of steam, the engine, on its trial, made 24½ revolutions per minute, driving a double acting pump 20 inches in diameter and 4 feet stroke. It raised 3,072,656 ale gallons of water to a height of 102 feet in twenty-four hours, corresponding to 35 effective horse-power. This engine worked expansively. In 1828, Cadwallader Evans, continuing his father's practice, erected an engine at Harrisburgh, Pennsylvania, in which he connected the governor directly with the expansion gear; but the steam valves were of the poppet variety.

cylinder at the same time; but nevertheless, in 1836, Pambour, in the first edition of his work, gave an accurate diagram for setting the eccentrics, and, except that he did not allow for the obliquity of the connecting and eccentric rods, he showed exactly what was the amount of expansion due to a lap so short as ¼th of an inch. The effect of lead had already been observed, and Pambour, who employed a lead varying between ⅛th of an inch, ⅜ths of an inch, and ⅝ths of an inch, in his experiments of 1834, described in 1836 the precise action of the slide valve, with both a definite lead and lap.

The first use of the lap valve in locomotive engines has been more commonly referred to the practice of the engineers of the Liverpool and Manchester Railway in 1838 and 1839. It is established, however, that the addition of lap was made in this case with but a partial, if not an erroneous, view of its real effect. So late, indeed, as 1844, the former chief engineer of the Liverpool and Manchester Railway read a paper before the Liverpool Polytechnic Society,* wherein he attributed nearly the whole advantage of lap, not to expansion, but to an earlier escape of the exhaust steam. Now, except as preventing back pressure against the pistons, it cannot be desirable to release the steam pressing upon them, until the full stroke has been completed. Yet, in the paper referred to, no experiments were detailed to show what the back pressure was, although it was inferred that, in certain instances, its resistance was equal to that of all the other resistances against which the engine worked. But even with the valves without lap it appeared that when, as was sometimes the case, ⅜-inch lead was employed, the ports were wide open to the exhaust when the piston had moved through only one-tenth of the return stroke, and this is more than can be said of the slide valves now worked by the link motion, especially on the higher grades of expansion. The author has had occasion to apply the indicator to engines with the least amount of lap; and even at high speeds the back pressure, although considerable, is not nearly so great as was counted upon by the Liverpool and Manchester Railway engineers in 1839. With valves having a travel of 3 inches, and without lead, a lap of ¾ths of an inch necessarily cuts off the steam at three-fourths of the stroke; yet, if there be no condensation in the cylinder, the steam admitted thus far exerts a mean force, throughout the whole stroke, equal to nearly 96 per cent. of what would have been obtained with an admission of steam during the whole stroke. The saving thus effected by the expansion due to a ¾-inch lap, on a valve moving 3 inches, is therefore equal to nearly 22 per cent. of the steam and fuel expended when working without expansion. It appears that when the lap was only nominal, or say ¹⁄₁₆th of an inch, the engines, under a careful system of firing, burnt 40 lbs. of coke per mile. With a ¾-inch lap the average consumption under the same circumstances as before was 32 lbs. per mile; and as the saving was here 20 per cent. of the original expenditure, it is evident that it might be fairly attributed to expansive working alone. Indeed, there are many engines now working without lap on the valves, but with separate expansive gear, and in which the consumption of fuel is no greater than in engines performing the same service with lap valves; notwithstanding that, in the former case, the steam is not permitted to escape from the cylinder until nearly the end of the stroke of the piston.

The real action of slide valves in locomotives, with respect to the movements of steam through the cylinders, could not possibly have been known until the application of the indicator. This instrument forms a court of appeal, wherein the speculative doctrines of the early engineers are controverted beyond dispute. It has shown that the chief purpose of lap, as applied to slide valves, is expansive action, and that a free exhaust of the steam which has already done its work in the cylinder is to be obtained by large ports and a quick action of the valve. It is the fact, however, that the consumption of coke in the Liverpool and Manchester passenger engines averaged 49 lbs. per mile in the summer of 1839, and that the consumption, after certain improvements made in 1840-42, was only 15 lbs. per mile. The whole saving has been often, but most erroneously, attributed to the addition of lap to the slide valves. For with the old valves the consumption of coke was diminished from 49 lbs. to 40 lbs. per mile merely by a better system of coke delivery; and even with 1-inch lap the minimum average consumption under the same system was 28 lbs. per mile. The whole subsequent reduction to 15 lbs. per mile, in engines of the same dimensions and performing the same work, was effected by increased care in firing, and more especially by greater accuracy of construction in the engines themselves. The last-named improvement was greatly due to the late Mr. Roberts. The saving effected by a 1-inch lap, as compared with no lap, never exceeded one-fourth of the original consumption of coke. All this had been before ascertained by engineers who, through lap on the valve, had sought to cut off at half stroke, and who had subsequently given up the lap valve, notwithstanding its economy in fuel, because it prevented the exertion of the full power of an engine in starting a train.

As for four fixed eccentrics in locomotives (instead of the old system of loose eccentrics, reversed by a treadle upon the foot-plate), James and Long, in America, employed them, the former in 1829, the latter in 1833, and Norris afterwards retained them in his engines made in 1834. Their first employment in England is attributed to the Messrs. Hawthorn, in 1837. The four fixed eccentrics and the lap valve were not merely mechanical improvements *per se;* but they were, still further, the foundations of most varieties of modern expansive gear, including the link motion, upon which, indeed, James had hit in 1829. In the earlier engines made by the American firm of Rogers, Ketchum, & Grosvenor, in 1837-38, the four fixed eccentrics were placed outside the driving wheels, in the manner subsequently adopted by Mr. Crampton.

The remaining improvement effected, in 1837, in the Locomotive Engine, was the introduction of counterweights in the driving wheels. The crank axles of the inside-cylinder engines, and the crank pins and bosses of the outside-cylinder locomotives—not to mention Forrester's cranks keyed to the axles outside of the frame—could not revolve from 100 to 250 times per minute without producing erratic motions of the engine upon the rails. This fact required but a very obvious induction for its comprehension, and it appears needless to attribute the "invention" of counterweights to any one. They were applied in 1837 by Richard Roberts, and also by Mr. Braithwaite on the (then) Eastern Counties Railway; and they were applied also in America in 1837 by Thomas Rogers and by George S. Griggs,—the last-named gentleman being probably the oldest American locomotive superintendent now living. The mechanical necessity which in very early times demanded the placing of winch-handles 180° apart in the circle of revolution required with greater urgency the application of counterweights to the rapidly revolving cranks of locomotive engines. Every engineer in the habit of reflecting for himself must have perceived this necessity for counterweights; yet it is the fact, as the author well remembers, that even later than 1840, and although it had been found necessary in practice to equilibrate locomotive driving wheels, these counterweights were looked upon as excrescences. They were seldom introduced into a drawing, and, with artistic prejudice, they were studiously excluded from published engravings. Probably the finest engraving yet made of a locomotive engine was that executed in 1847, by the late Mr. Gladwin, of one of Mr. Roberts' locomotives, as then constructed by Messrs. Sharp, Brothers, & Co., of Manchester. Yet, notwithstanding that Mr. Roberts first applied counterweights to his engines in 1837, the engraving in question does not show them. It was the custom at first to counterweight the wheels only to the extent of compensating for the weight of the revolving disturbing weights, making no provision for the momentum of the reciprocating parts. The quick reciprocation of a heavy piston and rod, with a cross-head and connecting rod, could not fail, however, to produce unsteadiness; and it became at length apparent that this could be corrected, to some extent at least, by revolving counterweights. The reform in this latter respect has been comparatively recent, however, and a notice of it would be premature in this place. The advantage of counterweights was less evident in inside-cylinder than in outside-cylinder engines. The former could be run with tolerable steadiness without them; but with the greater width between the lines along which the power is intermittently exerted in outside-cylinder engines, a compensation for the disturbing weights was indispensable to steadiness. But for counterweights, indeed, the outside-cylinder engine would have hardly been tolerated, much less widely preferred, as it now is, to the inside-cylinder arrangement.

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* Vide *Practical Mechanic and Engineer's Magazine*, vol. iv., p. 131.

The progress of improvement in locomotive workmanship was such, in the five years subsequent to the date of Pambour's first observations upon the Liverpool and Manchester engines, that Bury's locomotives upon the Grand Junction Railway cost but 2·73*d.* per train mile for repairs[*] for a service of upwards of 2,000,000 miles, in the years 1839 and 1840. The consumption of fuel was still large, however. In the first half of the year 1839 Bury's engines, just referred to, consumed 37·78 lbs. of coke per mile with passenger trains, and 43·39 lbs. with goods trains. With coke at even 20*s.* a ton,[*] however, the first-named rate of consumption corresponded to less than 4*d.* per train mile, and the last-named rate to less than 5*d.*, or considerably less than one-half of the previous cost of repairs alone on the Liverpool and Manchester engines. It was more especially in the quality of their workmanship that the engines of 1830-37 needed improvement, and to this improvement the practice of Mr. Roberts greatly contributed. The conditions of locomotive engine economy were, moreover, better understood a quarter of a century ago than is commonly believed at this time; and the specifications of the patents of this mediæval period of railway engineering attest the enlarged knowledge and abundant ingenuity which then prevailed. Roberts' variable expansive gear of 1832 was a striking instance of this. Perkins[†] not long afterwards embodied his well-known ideas of high pressure and expansive working in a description of a locomotive, to carry steam of 200 lbs. pressure, to be cut off at one-eighth stroke. In the same specification Perkins described the form of spark arrester now employed for wood-burning engines. Mr. Bodmer[‡] had still earlier proposed a variable chimney draught (not, however, by a variable adjustment of the blast nozzle); and he had described also the auxiliary steam blast or "blower" in the chimney, now so generally employed in coal-burning engines. Mr. Dodds,[§] too, had foreseen the disadvantage of the great friction of large slide valves under high pressure, and had specified an arrangement (now in favour on some of the French railways, and known as Plainemaison's valve) for placing the slide valve in equilibrium. In 1839, also, Mr. Dodds patented a three-cylinder locomotive[‖] and his wedge-motion eccentric. The latter, which will be more fully described on another page, was intended as a reversing gear only, if we may judge from the absence in the specification of any provision for lap or lead on the valve; but, by a proper adjustment of Dodds' wedges and eccentrics on the driving axle, the purpose of the "link motion" is now effected in many modern engines simply and successfully.

It may be right to mention here the progress of American practice down to 1840. The patent records form the most indisputable points of reference, and in many cases they anticipate the improvements attributed by tradition to those who were perhaps all the better engineers for not having been patentees. E. L. Miller, in June, 1834, and George E. Sellers, in May, 1835, obtained American patents for connecting the engine and tender so that a portion of the weight of the latter should be drawn by the former, so as to add to the weight on the driving wheels. This plan of increasing the adhesion of the engine in proportion to its load was long followed by the American makers, and in many cases it served to account for authenticated but otherwise inexplicable feats of traction. Until 1837 it was the custom of one American maker to employ only one pair of driving wheels, and to place these, as in the English engines, in front of the fire box. Another maker placed the single pair of drivers behind the fire box. Henry R. Campbell secured a patent, February 5, 1836, for combining with the ordinary truck or bogie two pairs of coupled driving wheels, one pair being in front, and one behind the fire box, as in the ordinary coupled passenger engine of the present day. The English goods engine of 1836 had six wheels, four being coupled in front of the fire box. In 1837, Eastwick and Harrison, of Philadelphia, added compensating levers, so placed that the weight of the eight-wheeled engine was carried upon four points upon the framing. This engine, shown in Fig. 54, may be considered as the first of a class which has since been generally adopted not only in America, but on the Continent of Europe, and to a considerable extent in Great Britain. The engine shown in Fig. 54 was first worked on the Reading Railroad early in 1839. It had 12-inch cylinders, 18-inch stroke, and four coupled driving wheels 3 feet 8 inches in diameter. Its whole weight was 12 tons 13 cwt. 1 qr., of which 8 tons 1 cwt. 2 qrs. were available for adhesion. With steam of 90 lbs. pressure per square inch (then a common pressure in American locomotives), Harrison's engine drew a load of 265 tons, including the tender, up a gradient varying from 27 feet to 35 feet per mile. The speed was not given, but the other particulars are derived from the report of a Committee of the Franklin Institute, dated May 9, 1839. Mr. Andrew M. Eastwick, one of the builders of this engine, had, on the 21st July, 1835, patented a mode of reversing locomotives, and which was employed in the engine whose performances have been just recorded. Instead of reversing the eccentric or the valve, a block was so made and placed between the valve and cylinder face that the steam ports might be made to communicate with either end of the cylinder at will. In Fig. 54 two valve spindles are shown entering the slide-valve box. The upper one is the working-valve spindle, and it is driven from a fixed eccentric on the hind driving axle. The spindle entering the lower part of the valve box is fixed within to the adjustable valve seat (the arrangement of which is shown in Fig. 55), and the other end of this spindle is connected to a reversing lever on the foot-plate.

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[*] *Vide* Mr. Bury's paper "On Locomotive Engines," Weale's *Quarterly Papers on Engineering*, vol. iii., pp. 306 to 310.

[*] The actual average price, according to Mr. Wishaw, was 37*s.* 4½*d.* per ton.

[†] Jacob Perkins' Specification, No. 7,069.

[‡] J. G. Bodmer's Specification, No. 6,616.

[§] Isaac Dodds' Specification, No. 6,470.

[‖] Dodds and Owen's Specification, No. 8,219, September 16, 1839.

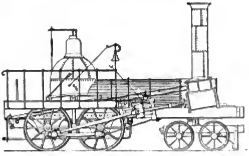
Fig. 54.—Eastwick and Harrison's Engine, 1837.

In Fig. 55, E is the slide valve and F the adjustable seat or block. When the latter is in the position shown in the middle figure, the front edge of the valve E admits steam to the front end of the cylinder. By moving the block F a short distance, however, the steam admitted under the front edge of the valve E will be conducted by another passage formed in the block F to the back of the cylinder. In conjunction with the Messrs. Winans, Mr. Harrison afterwards constructed and worked the rolling stock of the St. Petersburg and Moscow Railway; and all the engines, nearly 200 in number, originally made at Alexandrowski, near St. Petersburg, were fitted, and are still working, with Eastwick's reversing valve seat, with the addition, however, of a separate expansion valve. While mentioning Mr. Joseph Harrison, who was the original designer of the Norris engine, long made in America, it may be added that, besides now being one of the most successful and influential engineers in his own country, he is the patentee of the form of cast-iron boiler which is now attracting so much attention in Lancashire.*

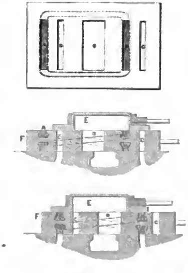

Fig. 55.—Eastwick's Reversing Valve Block, 1835

One of Norris' engines, designed by Mr. Harrison, was tried in July, 1836, on an inclined plane, rising 1 in 14, or 369 feet per mile, on the Columbia Railroad, near Philadelphia. Its own weight was 6 tons 8 cwt. 2 qrs., and it drew an additional load of 8 tons 11 cwt. 2 qrs. up the plane 2,800 feet long. This feat was represented as having been performed with steam of 60 lbs. pressure. It will be easily seen, however, that with 10-inch cylinders, 18-inch stroke, and 4-feet driving wheels, it could not have been accomplished with less than 90 lbs. of steam, especially as the run was made at the rate of rather more than 15 miles per hour. This, however, and other performances of the Norris engines, induced Captain Moorsom to order a number of them for the then Birmingham and Gloucester (now part of the Midland) Railway. The first of this order was tried in April, 1839. The engine had outside cylinders 10½ inches in diameter, 18 inches stroke, a single pair of 4-feet driving wheels, and a truck of four wheels 2 feet 6 inches in diameter. The boiler had seventy-eight copper tubes 2 inches in diameter and 8 feet long. The weight in working trim was 9 tons 11 cwt. 1 qr., and that of the tender with its ordinary charge, 6 tons 4 cwt. 1 qr. In working, however, a part of the weight of the tender was drawn upon the driving wheels, in the manner already mentioned in connection with Miller's and Sellers' patents. It had been stipulated in the contract that, with steam of 60 lbs. pressure, the Norris engines should draw 100 tons gross at the rate of 20 miles per hour up inclines of 1 in 330, and the same load at 14 miles per hour up 1 in 180. On trial, the prescribed load was taken up an incline of 1 in 177 at a rate varying from 9·8 to 13·8 miles per hour, thus falling but a little short of what had been promised. These engines (twelve or more were ordered) worked with an extravagant consumption of coke in proportion to the water evaporated. Captain Moorsom stated to the Institution of Civil Engineers that in a lengthened period of service they evaporated 25,084 gallons of water with an expenditure of 26 tons 11 cwt. of coke, corresponding to less than 4½ lbs. of water per pound of coke. This waste was doubtless due to the sharp blast employed in overworking a small boiler, although the indicated power could hardly have exceeded 90 horse-power. These engines, with their tenders, and including a 20 per cent. import duty, cost from £1,500 to £1,600 each delivered on the line.

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* A communication to J. H. Johnson, August 30, 1859, No. 1,970. See also *Proceedings of Institution of Mechanical Engineers* for 1864; also *The Engineer*, vol. xvii., p. 291.

Before dismissing the Americans for the present, it may be added that they were among the earliest to work steam expansively in locomotives, although expansion was not effected in the Norris engines imported into this country. In addition to the early instances of expansive working already mentioned, an engine was fitted with separate expansion valves by David Matthew, in 1837, from which time like gear was regularly employed on the Utica and Schenectady Railroad. On the 17th of May, 1838, Isaac Adams patented the application of an expansion valve on the back of the main slide valve, steam ports being made through the latter, as shown in Fig. 56.

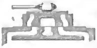
Fig. 56.—Adams' Expansion Valve, 1838.

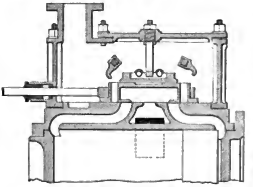
Fig. 57.—Edwards' Valve, 1840.

Adams' expansion valve was worked by an eccentric, and the arrangement was for a long time employed by the American makers. M. Farcot, in France, had in 1836 employed a gridiron expansion valve on the back of the main valve; but the former moved with the latter until its position was reversed by striking fixed stops. A later application of this system of expansion-valve stops was that of Edwards, employed on the French railways about 1840, and shown in Fig. 57.

In August, 1841, Horatio Allen, of New York, patented means whereby Adams' valve could be adjusted at will to cut off at various parts of the stroke. This was effected by making the expansion valve in halves, which could be separated or brought nearer together by means of a right and left-handed screw, as in Fig. 58.

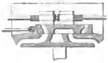
Fig. 58.—Allen's Expansion Valve, 1841.

This arrangement was afterwards employed in England by Mr. Wilson (of Messrs. Nasmyth, Gaskell, & Co.), and by M. Meyer on the Continent. It may be mentioned here that Ross Winans patented an arrangement, July 29, 1837, by which the same slide valve might be worked either by an eccentric set for nearly full stroke, or by one or more cams formed and adjusted to cut off at any desired part of the stroke, both the eccentric and cam being employed in the same engine. This valve gear was employed by Mr. Winans in a large number of locomotives, and in a practice of nearly twenty-five years. The various American patents referred to are described in the *Franklin Institute Journal*.

In even the lengthy summary of locomotive history with which we have now reached the year 1840, the works of a few engineers have been overlooked. It might have been mentioned, on a previous page, that the first locomotives made in Scotland are believed to have been two constructed in 1831 by Messrs. Murdoch & Aitken, of Hill Street, Glasgow, for the Monkland and Kirkintilloch Railway. These engines, designed by Mr. Dodds, were of nearly the same construction as the Killingworth engines, with the exception that they had multitubular boilers. These engines, the first of which began to work May 10, 1831, had 10½-inch cylinders and 24-inch stroke, and they worked with steam of 50 lbs. pressure. Their ordinary speed was 6 miles an hour; at which rate they drew 200 tons on a level, and through curves of a radius of only 340 feet, or rather more than 5 chains. A drawing of one of these engines is given in Hebert's *Encyclopædia*.

In 1833, Robert Stephenson patented the application of tyres without flanges to the driving wheels of six-wheel engines—the driving wheels, in this case, being, of course, between the leading and trailing wheels. With plain tyres it thus became practicable to work engines, with a wheel base of 10 feet or 12 feet, around the sharpest curves then permitted upon English railways. In the same specification Mr. Stephenson described the steam brake; but this had been patented a year earlier by Mr. Roberts. It may be mentioned here that there is a patent in the name of Robert Stephenson, and dated 23d January, 1826, for providing a separate axle for each wheel of railway carriages, an arrangement permitting each wheel to turn independently of the others. The same arrangement was adopted in 1860 by M. Flachat, for the rolling stock of a proposed railway over the Alps; and it has been since re-patented by Messrs. Samuel and Train. The original invention was not by the late Mr. Stephenson, M.P., who, in 1826, was absent in South America, but was patented by his uncle, for whom he appears to have been named.

Mr. Carpmael patented a locomotive engine in 1835, in which the steam was worked expansively, by giving to the slide valve two distinct and successive movements at each stroke of the piston. This plan, as has been seen, had been adopted by the French engineers at an early date, and it had been proposed by Tredgold in 1828. It was, as has been mentioned, afterwards adopted by Ross Winans in the United States. There was at one time a dislike to the eccentric as a means of transmitting motion to the slide valves of quick-working engines. Its action, in opening and closing the valve, is indolent, when the necessity for an instant admission and exhaust of the steam is considered. In White's *New Century of Inventions* this objection is raised, and a cam motion described as a substitute for eccentric gear. But although there are instances of the continued working of cam valve gear at considerable speeds, the continuous motion of the eccentric has been found indispensable to durability.

In 1837, Dr. Church, an amateur engineer at Birmingham, constructed and worked a four-wheeled tank engine, with horizontal outside cylinders 11¼ inches in diameter, 24-inch stroke, and a single pair of driving wheels of the large diameter of 6 feet 2¼ inches. The whole weight was 14 tons in working trim, two-thirds of the weight being on the driving wheels, which were placed nearly under the middle of the length of the boiler, the cylinders being behind the fire box. The boiler was of the usual kind, except that it had the large hemispherical dome over the fire box, at that time employed by Bury and by the American makers. The boiler exploded, however, and Dr. Church was killed. His engine, the "Eclipse," was doing duty in 1861 at Swansea. Mr. F. W. Turner,* who reconstructed it there, found it had hollow connecting rods, gradually tapering from the middle towards each end, as in the ordinary solid rods. The piston rods were guided by wheels working on guide bars, as in Maudslay's "table engine" (originally designed by Mr. Robinson Kittoe), and the valves were pistons working in an open pipe. The engine was fitted, in 1857, with six coupled wheels for coal traffic. It may be added that the tubes of the original engine were fixed by an expanding mandril,† invented by Dr. Church in 1833, and used, to his order, by Mr. John Inshaw, of Birmingham. In 1861 the original mandril was for some time in the keeping of the author of the present work.

The valve gear of some of the early engines will be described in a future chapter, but it may be added that, after the adoption in England of four fixed eccentrics, the reversing gear employed was often of a most complicated character.

Attempts were occasionally made, between 1833 and 1840, to burn coal in locomotive engines without the production of smoke. John Chanter, in 1834, patented a furnace in which the coal was fed through a hopper, upon a "distilling plane," and the gas, as it was expelled, was met by highly heated air. The under side of the crown plate was covered by firebricks, and a deflecting bridge of firebrick lumps, or tiles, was to be employed. The chemical knowledge displayed in Chanter's specification is sound, and his views might be studied with profit by many who have since turned their attention to coal burning. Chanter's patent of 1834 referred to the furnaces of land and marine engines; but he subsequently became associated with John Gray in a series of patents for the most extraordinary forms of coal-burning fire boxes for locomotives. One of these was tried on the Liverpool and Manchester Railway, but with results inferior to those attained with the ordinary construction of fire box. In 1837, Mr. Booth, the treasurer of the Liverpool and Manchester Railway, patented a coal-burning locomotive fire box, in which the long and steeply inclined grate now employed on the South-Eastern, Great Northern, and other lines, was introduced. The trailing axle was beneath the grate, and a feeding box for coal was placed at the back. The door was of the sliding variety, and was much the same as that now known as the "Midland door," and shown in Plate IV. It does not appear that Mr. Booth's boiler was ever tried.

From the period of the first use of locomotive engines, their size and weight have been almost constantly increasing. They had attained to respectable dimensions by 1840; for after the success of railways had been so fully proved, it was sought to move heavier trains at higher speeds, and the consequent necessity for enlarging the locomotive engine gave considerable scope to the labours of engineers. Indeed, a certain class of engineers soon began to complain of the insufficiency of the ordinary 4 feet 8½ inch gauge; and the fact that at one time there were no less than seven different widths of railway gauge in the United Kingdom, as there now are in the United States, was due in some measure to the impatience of locomotive makers to expand the sizes and amplify the arrangement of their engines to meet the growing necessities of railway traffic. The 7-feet gauge was not, however, originally worked with heavy engines, the only parts which were notably enlarged being the driving wheels. But the London and Birmingham and Great Western Railways had very favourable gradients, on which a 12-ton engine could perform as much work as a 20-ton engine on a line having frequent inclines of say 1 in 100. Down, indeed, to 1845 the London and Birmingham engine stock included nothing of more than 12½ tons weight; and of ninety engines, all but one were on four wheels. The passenger engines had 12-inch cylinders, 18-inch stroke, and 5½-feet wheels; while the goods engines had 13-inch cylinders, 18-inch stroke, and four coupled wheels of 5 feet diameter. The passenger engines weighed 10½ tons in working order, and with a total heating surface of 420 square feet, they evaporated 75 cubic feet of water per hour, when working at a mean speed of 30 miles per hour. In some cases 80, and in one case 91, cubic feet of water were evaporated per hour by these engines. They worked trains of 42 tons' weight at an average speed of 25 miles an hour, and in some cases moved trains of 37½ tons at a speed of 50 miles per hour. One of the best performances of this class of engines (with 5-feet driving wheels, however) was that obtained on the Grand Junction line by Dr. Lardner, who accompanied one from Liverpool to Birmingham and back, a distance of 190 miles, taking a train of 12 carriages, weighing 60 tons, besides the engine and tender of 22 tons. The mean speed, including four stops each way, was 29¼ miles an hour, the consumption of coke being 37·1 lbs. per mile, and the evaporation 91 cubic feet per hour, and 5·7 lbs. of water per pound of coke. The rate of evaporation, per unit of heating surface, was about the same as in the earlier Liverpool and Manchester engines. The actual resistances overcome cannot now be known, but later experience will justify an estimate of 25 lbs. per ton of engine, tender, and train, a portion of this being allowed for gravity on inclines. Taking this as the resistance, the engine would have exerted 164 indicated horse-power, corresponding to 6⅔ lbs. of coke and 34·6 lbs. of water per indicated horse-power per hour. It is not probable that the engine (the "Hecate") which gave these results had lap valves or expansion gear, and its performances must therefore be reckoned as very creditable under the circumstances.

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* Vide *The Engineer*, vol. xi., p. 361.

† Vide *The Engineer*, vol. xii., p. 77.

The lines with steeper gradients were worked with heavier engines. As a class, the engines of the Manchester and Leeds Railway were the heaviest, in 1839, on the narrow gauge. They had 14-inch cylinders, 18-inch stroke, and 577 square feet of tube surface. Their weight, empty, was 15 tons, and loaded, about 17 tons. As early as 1835, Messrs. Tayleur had made an engine, for the Leicester and Swannington Railway, with 16-inch cylinders, 20-inch stroke, and six coupled wheels of 4½ feet diameter. The boiler was 4 feet 1 inch in diameter. Messrs. Stephenson soon afterwards made an engine of the same general dimensions for the same line. The boiler, however, was 3 feet 11 inches in diameter, and 8 feet 6¼ inches long, and contained 154 tubes 1⅝ inch in diameter, presenting 589 square feet of heating surface. The fire grate was 10·15 square feet in extent, and the area of the fire box was 67·45 square feet. The steam ports were of the then large size of 8¼ inches by 1⅝ inch.

Besides the large driving wheels of the Great Western engines, the North Midland engines, made in 1838, had 6-feet driving wheels, then the largest on the narrow gauge, although the same diameter had been occasionally employed in earlier engines. A large number of the engines made in England in 1838-40 for the Continental railways, had 6-feet driving wheels. Among these were the engines of the Vienna Railway, Taunus Railway, Paris and Orleans, Berlin and Saxon, Berlin and Potsdam, and other lines.

Previous to 1840, Messrs. John and George Rennie had made two engines for the Croydon Railway with outside horizontal cylinders, the connecting rods working upon pins in the driving wheels. These engines had four coupled wheels 5 feet in diameter, besides a pair of smaller trailing wheels. The principal pair of driving wheels had cylindrical tyres without flanges. The valve motion was of the variety known as Carmichael's, with a single fixed eccentric for each valve.

Down to 1840 the working pressure in locomotives had seldom been increased beyond 50 lbs., only a few engines working at 60 lbs. By Mr. Wishaw's elaborate reports of his examination of the railway working of the period, it appears that high rates of speed were maintained. Occasional instances of 50 miles an hour were to be met with, and several of the Grand Junction engines, both with and without trains, were sometimes permitted to descend the Whitmore and Madeley inclines at a rate, for short distances, of 68·18 miles per hour!

It will be right here to notice the anxiety among the earlier locomotive engineers to keep down the velocity of the pistons at a high speed of train. Mr. Brunel's engines, with 16-inch stroke and 8 feet wheels, were designed with reference to this object. An 8-feet wheel makes 210 revolutions in a mile; and at 50 miles an hour the piston driving it would, if the stroke were 16 inches, move at the rate of 466⅔ feet per minute. But there is no necessary speed of piston beyond which it has been found impracticable to work an engine; while, on the other hand, there is a certain amount of tractive force which is indispensably necessary to the draught of a particular load at a given speed. Now, with respect to the tractive force of an engine having wheels of a given diameter, and with steam of a given mean pressure in the cylinders, the area of the pistons must be increased exactly as the length of stroke is diminished. The progressive force at the periphery of the driving wheels is perhaps the first point to be taken in considering the power of a locomotive, although, more strictly, the real power is expressed by the product of the tractive force into the velocity of the engine. If we have to provide a tractive force of 2,500 lbs. at the periphery of an 8-feet wheel, and the length of crank be but 8 inches, we must have a pressure of 15,000 lbs. upon the crank pin; and as the crank pin always moves 1·57 times faster than the piston which drives it, the necessary pressure upon the piston would be about 23,550 lbs. If, however, the length of crank be 16 inches, but one-half this pressure is necessary; while it would appear more judicious to exert this diminished pressure in an engine with 24-inch stroke and 6-feet wheels, wherewith (other things being equal) the tractive force and the speed of piston at any given number of miles per hour would be the same as with a 32-inch stroke and 8-feet wheels. For a given tractive force a 24-inch stroke and 6-feet wheel would require but one-half the total pressure upon the pistons which would be necessary with a 16-inch stroke and 8-feet wheels. But at the same time the speed of the piston would be twice as great, or at 50 miles an hour, 933⅓ feet per minute; which, indeed, is now found to be a very good speed for a locomotive piston. The early Liverpool and Manchester engines, with 18-inch stroke and 5-feet wheels, worked at a velocity of 504 feet of piston per minute when running at 30 miles an hour; and when, as was occasionally the case, 40 miles an hour was reached, the rate of piston was 672 feet per minute. We have already seen how an engine with 18-inch stroke and 5½-feet wheels attained a maximum speed of 50 miles an hour; and here the velocity of the piston was 765 feet per minute. Yet the prevailing opinion among engineers a quarter of a century ago was that the nearer all engine pistons could be kept to the usual rate of Boulton & Watt's engines—say 220 feet per minute—the better. There are many, indeed, who still prefer this rate of speed; and it may be at once admitted that high piston speeds do require increased wearing surfaces and better fitting than would answer in one of the old pattern of condensing engines.

There is still a disposition to employ large driving wheels for fast engines, even with a stroke not exceeding 24 inches. The heaviest broad gauge passenger engines have 18-inch cylinders, 2-feet stroke, and 8-feet wheels, although, with a speed of piston one-third greater, precisely the same tractive force would be exerted, at the same mean pressure per square inch upon the pistons, with 15⅜-inch cylinders, 24-inch stroke and 6-feet wheels. The large cylinders and large wheels are occasionally to be found upon the narrow gauge, where, with a boiler of given size, the smaller cylinders and smaller wheels would afford equal tractive force, at the same speed of train, and obviously with less weight in the engine. In most cases, indeed, where large wheels have been introduced upon the narrow gauge, it has been found expedient to cut down their diameter, 8-feet wheels having in many cases been brought down to 6 feet, and 7-feet wheels to 5½ feet. It is only at the very highest rates of speed that diameters of more than 6½ feet now appear to be necessary.

CHAPTER III.

HISTORY OF THE LOCOMOTIVE ENGINE—*continued.*

The general adoption of variable expansion gear in locomotive engines dates from about 1845, although the link motion, first worked out by James and Dougherty in 1832, was introduced into Messrs. Robert Stephenson and Co.'s engines in 1843. So far as James was concerned, his link motion perished with the explosion of his engine; and the re-appearance of this princely valve gear is in no way referable to his unlucky labours. Nevertheless, as in America, so in England, the link motion—the adoption of which marks an era in locomotive history—was not the direct result of original invention, but rather the fruit of a pregnant practice from which all the separate features of this favourite gear were inherited. The provision of lap on the valve was the first step, and the adoption of four fixed eccentrics the next; although there were other stages in the progress towards completion. When page 49 of the present work was written the author was not aware how extensively lap had been adopted in the valves of the early engines. Its presence might, indeed, have been directly inferred from the sketch given in Wood's treatise *On Railroads* (Fig. 59) of the eccentric motion for working the valves of the Killingworth engines.

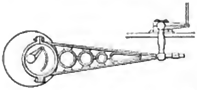

Fig. 59.—Valve Motion of Killingworth Engines, 1829.

Here, although the valve has an intermittent motion, produced by the striking of a collar on the end of the eccentric rod against fixed stops, the form of the curved slot in the eccentric defines the position of the latter in forward and back gear respectively. There must have been a great amount of lead on the valves, or else considerable lap. It is of course possible too, as was the case with nearly all the illustrations of Mr. Wood's book, that the valve gear in question was not accurately drawn. But apart from the instances of the early adoption of lap, as given on a previous page, it appears that its effect was well known in the north country locomotives of 1829 and 1830. Mr. John W. Hackworth, of Darlington, informs the author that Timothy Hackworth's engine, the "Sanspareil," of 1829, had about half-inch lap on the valves, and that in 1830 he adopted a proportion of lap to travel of valve which gave an admission of steam for about 85 per cent. of the stroke, leaving 15 per cent. to be worked expansively. The present Mr. Hackworth, stating this in his own way, says—"My late father established a system in 1830 (just after the 'Globe' engine) which he practised for some years, viz., to steam on an average 135°, working the remaining 45° expansively in each half revolution, being a little less on one side and a little more on the other, owing to the practical inaccuracy of the gear. I believe the 'Majestic' engine, the first of six all alike, had this first applied (speaking from memory) about the latter end of 1830 or the beginning of 1831. In 1835 the 'Magnet' engine was made to work a little more expansively than 45°" (15 per cent. of the stroke).

In Mr. Roberts' prescient specification of 1832, the valve shown in Fig 40, page 40 of the present work, was described, and but from one-eighth inch to one-fourth inch lap was stipulated, this being obviously as an assurance that the valve should be steam-tight at half throw. The effect of lap was secured, however, more especially in the supplementary expansion valve, by working that valve so that the steam slots cut through its sides should work alternately more or less beyond the ports with which they were in adjustment. Steam was cut off in either direction at the moment when the slots in the valve passed completely beyond the steam ports, as in the ordinary gridiron valve; and if, by so increasing the motion of the valve by the variable gear proposed by Mr. Roberts, the steam ports were thus kept closed during a considerable part of the whole motion of the valve, the period during which they were thus closed represented a corresponding expansion within the cylinder. Mr. Roberts' complete knowledge of the *rationale* of variable expansion gear appears from his own words. He says in his specification (October 11, 1832)—"The induction valve makes a single stroke by a half revolution of the eccentric which works it; consequently, whether those strokes be long or short, they are each performed in an equal interval of the engine's general action. According, then, to the length of range of stroke of the induction valve, the steamway into the work valve cylinder is kept open a longer or shorter portion of the whole duration of each stroke. Now, the extent of range of the induction valve is governed under the arrangement now described by the variable length of the curved arm of the bent lever K, the range of the valve being extended as that arm is lengthened. The proportion then betwixt the axual extension of the steam passages and the longitudinal range of the valve determines the relative times of free and closed passage for the steam. If the valve move through somewhat less than twice the axual extension of the steamway, the passage for the flow of steam will never be cut off; if a longer stroke be made by the valve it will close the steamway during the latter portion of the stroke of the piston; and if the range be further increased, the steamway will be closed during a proportionately longer portion of the stroke. It is desirable to make the adjustment such that the valve will cut off at one-eighth or one-sixth of the stroke by its longest action, and keep the steamway constantly open by its shortest. I find that if the shortest action of the valve be made about one and a half times the axnal length of the steamway, and the longest about five times, the variability is sufficiently great. In the common working valve the action of the eccentric is about one-fourth later in time than the engine crank; in my present arrangement the eccentric for the induction valve is about one-third later than the crank. The induction valve must be so adjusted as to be just commencing to open the steamway, H, as the piston is finishing its stroke."

We have given this and other extracts from this remarkable specification of Mr. Roberts (and it was drawn by his own hand) as containing the earliest statement of the principle of, and the most complete contemporary means for effecting, variable expansion. The "induction valve" referred to was the supplementary expansion valve, the "work valve" serving the purpose of the ordinary slide valve without lap. The late Mr. Roberts did little to carry the ideas embodied in this specification into practice, his attention being much engrossed between 1832 and 1840 with the perfection and manufacture of his self-acting spinning mule. Nor did he set especial value upon the mechanical prevision and sound reasoning to which his specification of 1832 testifies; for in the author's lengthy and intimate personal acquaintance with Mr. Roberts, that gentleman never spoke of his variable expansion gear until when, less than a month before his death, the author having met with this patent in the course of a laborious search, questioned him concerning it.

It should be mentioned that, as early as 1828, M. Hallette, of Arras, and who was at that time one of the leading mechanical engineers in France, employed variable expansion gear in the light draught steamboats constructed by him for navigating the Garonne. M. Hallette patented his steamboat machinery in December, 1828; and in the *Brevets d'Invention* (vol. xxxviii., page 332 and plate xxii) is shown a separate expansion valve working in a separate case at the back of the main valve. The means for adjusting the expansion valve are not clearly shown; but in this specification, which is unusually complete, it is stated that in actual practice the steam was cut off variously at one-third, one-half, and three-fourths of the stroke.

The link motion was not directly worked out, however, as the solution of the problem to which M. Hallette and Mr. Roberts had addressed themselves, but, as in James' case, it was first hit upon as a reversing gear, its capability of variable expansion, while preserving nearly constant lead, being, most likely, rather discovered than foreseen. With a single fixed eccentric, which admitted of little if any lap or lead, there was no difficulty in reversing. It is not certain who was the first to engage the end of one and the same eccentric rod with the ends of arms projecting from opposite sides of a rocking shaft, but Messrs. Carmichael, of Dundee, employed this plan in 1818 in the twin ferry-boats working across the Frith of Tay, and it is figured and described in Gregory's *Mechanics*. This general mode of reversing was for a long time in favour, having been adopted with various but immaterial modifications in the engines of Goldsworthy Gurney, Forrester, Rennie, Cavé (of Paris), and by Baldwin and Dunham in America. Fig. 60 shows the form employed by Cavé, the reversing arm from the weigh shaft having a roller at its end, working in a long slot in the eccentric rod. The gabs or hooks in the end of the eccentric rod could be readily engaged with either end of the rocking arms by which the valve was worked; and it is obvious that, if the gab were engaged with the end of the upper arm, the motion transmitted to the valve would be exactly opposite to that which would result from the motion of the eccentric rod when it occupies the position shown. It is easy to understand that it reqnired a further discovery to reconcile this reversing gear with the requirements of lap and lead on the valve. At all events, it was necessary for this purpose that the eccentric rod should be shortened, or the rocking arms proportionately lengthened, beyond the proportions adopted by Messrs. Carmichael. Fig 61 illustrates the effect which such alterations would have. Let A B represent the centre line of the engine, and the circle *a b* B the path described by the centre, *b*, of the eccentric. C is the rocking shaft, and *c* and c^1 the stud pins in the ends of the opposite arms projecting from this shaft. Let it be supposed that the slide valve derives its motion from the stud pin *c*. Noting the position of *c*, it

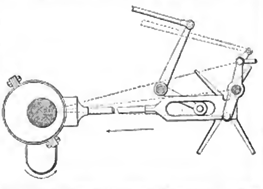

Fig. 60.—Cavé's Adaptation of Carmichael's Reversing Gear, with Single fixed Eccentric, 1835.

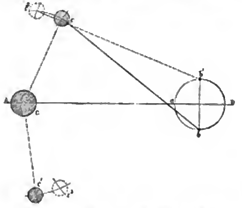

Fig. 61.—Carmichael's Reversing Gear, proportioned for Lap and Lead.

will be seen that when the centre of the eccentric, after half a revolution, arrives at b^1, the position of c will be changed to c^2. If the eccentric rod, b c, be now disengaged from c, and lowered to c^1, the valve will not be moved by this change, as long as the stud c^1 and the centre of the eccentric, b, occupy the position shown; but as soon as the engine begins to move, the motion of the valve will be opposite to that which would result if the eccentric rod were engaged with c. Thus the engine is reversed, as in Fig. 60, but the difference of position between c and c^2, and between c^1 and c^3, will correspond to twice the sum of the lap and lead of the valve, if worked from c. And it will be found that the angle made by the eccentric rod with the centre line of the engine, in top and bottom gear respectively, corresponds in effect with the same amount of angular advance given to the eccentric itself, when working the valve through a rod leading directly from the eccentric to the valve. Within reasonable limits the single fixed eccentric may, as shown in Fig. 61, work a valve with any required amount of lap and lead in both forward and back gear. Thus Carmichael's gear, properly proportioned, serves an important purpose in addition to that merely of reversing the engine. Its applicability to lap valves was not demonstrated in the ferry-boat of 1818, already referred to, but it is evident in the short eccentric rods and long rocking arms of Forrester's engine of 1834 (see Fig. 42, page 41 of the present work). The American builders, Baldwin and Dunham at least, adopted still shorter eccentric rods, and were thus enabled to give considerable lap to the valves. This lap is shown in the working drawings of Dunham's engine in the fourth volume of the *Practical Mechanic and Engineer's Magazine*, Glasgow, 1845. Although the total movement of the valve, as shown in these drawings, is but 2 inches, the valve has a lap of ½ inch at each end, equivalent to 1 inch lap with a 4-inch travel; and thus, even if there were no lead, it would cut off the steam when the piston had made three-fourths of its stroke; while with lead, the point of cut off would be correspondingly earlier. As a reversing gear, the Carmichael arrangement of a single fixed eccentric was simpler than the link motion, and it was incomparably simpler than most of the other contrivances for reversing which came into use after the general adoption of four fixed eccentrics. The single fixed eccentric could not, however, give variable expansion, and if the gab of the eccentric rod was made to bear properly upon its stud pin in forward gear, the opposite gab, when the engine was in back gear, could not well be kept up fairly to its corresponding stud pin, through the whole range of the motion of the latter. This will be evident upon an inspection of Fig. 60, where it will also be seen that, unless the roller in the end of the weigh arm is of somewhat less diameter than the width of the slot in which it works, the bottom gab would not bear properly upon its pin throughout the whole range of its motion. The Carmichael gear had also another fault. The mere disengagement of one gab did not of necessity engage the other, and hand gear was in many cases required, especially for locomotives, to get the valve over to its right place in starting. This was also necessary with the single loose eccentric, both that employed in the Killingworth engine, Fig. 59, and that subsequently used in the Liverpool and Manchester engines.

Fig. 62 shows the valve gear and mode of reversing employed for a time by the Messrs. Hawthorn, and in which no eccentric was employed. A block attached to the connecting rod at about the middle of its length, worked in a slotted frame, and from this, motion was taken off to the valve through a rocking arm like that employed by Carmichael. The slotted frame was, moreover, so adjusted, with respect to the centre line of the engine, that the advance of the valve corresponding to the lap and lead was given in both forward and back gear. This, and Sharp, Roberts and Co.'s reversing gear, with four fixed eccentrics, will show what contrivances were resorted to previous to the adoption of the link motion in 1843.

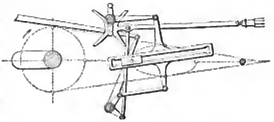
Fig. 62.—Hawthorn's Valve Gear, 1840.

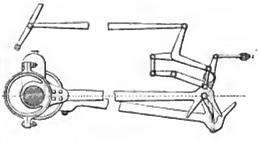
Fig. 63.—Sharp and Roberts' Valve Gear, 1842.

On the 26th July, 1838, John Gray, who was then connected with the Liverpool and Manchester Railway, patented a variety of schemes relating to locomotive engines. Among these was the form of valve gear shown in Fig. 64, and which was applied to the engine "Cyclops," at that time running upon the Liverpool and Manchester line. This gear bears upon the history of the link motion, inasmuch as Gray contended that it

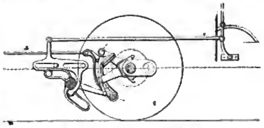
Fig. 64.—Gray's Expansive Gear, 1838.

was the best form of link motion, and he brought an action, some years afterwards, against the London and North-Western Railway Company for infringement of his patent, although the engines of that line then had the kind of link now in use. Gray employed a species of link, *a*, vibrating upon an axis at its lower end. By an elaborate arrangement of reversing gear the end of either eccentric rod could be engaged in its corresponding curved groove, there being two grooves on opposite sides of the link. The eccentric rods, moreover, could be adjusted to give their motion to the arm *a* at variable distances from the axis upon which it vibrated: *b* is the valve spindle, and *c* the reversing rod. This arrangement varied the travel of the valve, and it varied the lead at the same time, and where there was already considerable lap, the variation of lead proceeded more rapidly than the variation in the travel of the valve. With a lap of 0·7 inch, Gray states in his specification that, in increasing the throw of the valve from 2 inches to 4½ inches, the increase of lead was from ⅛th inch to 11⁄16ths inch. The whole arrangement was a clumsy attempt to apply Mr. Roberts' curved valve arm to work the main slide valve, and to combine with it the means of reversing. Gray's specification is an interesting document, although it contains no drawings, and notwithstanding that some of the inventions described in it are nearly or quite impracticable. One claim was for applying a trunk to one face of a locomotive piston, thereby diminishing the effective area of that face, and upon the diminished area steam was admitted directly from the boiler, while the steam so admitted was afterwards exhausted into the opposite end of the cylinder, to act expansively upon the full area. This plan has been often re-patented within the twenty-six years since Gray proposed it. Another claim of Gray's was to work engines by hot water, or rather, by the steam disengaging from it within the cylinder, the water being afterwards pumped back into the boiler, and the disengaged steam sent out into the atmosphere or into a condenser. Another claim was for turning eccentrics out of the solid driving axle of an engine, the axle being first turned to the full diameter of the eccentrics, and afterwards suitably centred and turned down between them. Another claim was for the combination of inside journals for the driving axle with outside journals for the leading and trailing axles—a plan much followed since, and which is embodied in Messrs. Beyer, Peacock, and Co.'s engine "Dom Luiz," shown in Plates III. and IV. of the present work. To such an extent did Gray's partiality for this arrangement go, that in some coupled engines he designed for one of the northern lines he had a crank axle not only for the driving wheels, but for the leading wheels also, in order that they might be coupled without interfering with the outside bearings of the leading wheels! Another claim was for a variable blast orifice, and the arrangement described possessed some merit; but Gray's object, as expressed by himself, was not to influence the rate of combustion by the varying intensity of the exhausting steam, but to check the velocity of the engine when required, by increasing the back pressure upon the pistons. This was Booth's idea in his patent for a variable blast, dated 1836, only Booth went so far as to propose, in some cases, to completely close the blast pipe.

Gray's last claim was for a sliding tube plate, working within a stuffing box in the front end of the boiler, although means were not pointed out for preventing the tendency of this plate to blow out under the pressure of the steam.

On the 16th September, 1839, Messrs. Dodds and Owen patented their wedge-motion eccentric, shown in Fig. 65. In this they were upon the threshold of the discovery of a useful and effective variable expansion gear; but, singularly enough, while they proposed to vary the motion or travel of the valve, they had not, apparently, thought how to turn this to account in expansive working. By a slight adjustment of the very parts proposed by Dodds, and without any real alteration in their form, they are now employed, in many cases, to produce all the effects of the link motion. The action of Dodds' gear may be understood from Fig. 66, in which three arrangements of it are represented. A B, a line common to the three, represents the direction in which the crank, in each case, is supposed to lie. The smaller circles represent the paths described by the centre of the eccentric in different positions with reference to the centre of the axle. The larger circles represent the eccentric itself, in full and dotted lines. In No. 1 the valve is supposed to have neither lead nor lap, and in full gear in one direction the centre of the eccentric is at *a*, the eccentric radius being at right angles to the crank. It is evident that if the eccentric can be so moved upon the axle as to bring the centre of the former to *e*, the motion then given to the valve will be opposite to that when the centre of the eccentric was at *a*. The engine will be thus reversed, the new position given to the single eccentric producing the same effect as that which would result from having another eccentric with its centre at *e*. So far as the effect upon the motion of the valve is concerned, it is here immaterial whether the eccentric is turned around upon the axle so as to bring the centre *a* to *e*, or whether the eccentric be moved directly across the axle in the line *a e*. If the eccentric can be adjusted only by turning it around the axle, as in the old Killingworth and the Liverpool and Manchester

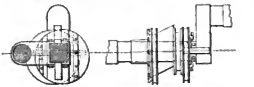

Fig. 65.—Dodd's Wedge Motion Eccentric, 1839.

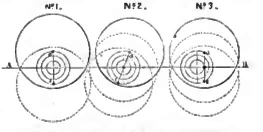

Fig. 66.—Diagrams of Action of Dodds' Wedge Motion

arrangement of a single loose eccentric with stops, the motion given to the valve cannot be varied. If, however, the eccentric be moved across the axle, and means be provided for securing it in different positions, say when it is one-sixth, one-third, one-half, or three-fourths across, as well as when it has been moved entirely across, it is evident that different amounts of motion may be thus given to the valve. In the diagram No. 1, Fig. 66, no motion whatever will be given to the valve when the eccentric is moved half-way across the axle, because the centre of the eccentric will then coincide with that of the axle; and thus, according to the adjustment of the eccentric, the valve may receive any amount of motion, from nothing to the full amount of eccentricity; and this range of variation is commanded in both forward and back gear. It was this range of motion which Dodds proposed to obtain by means of his adjustable eccentric; but when the valve has neither lap nor lead the only effect of diminishing its motion is to wire-draw the steam, and when the eccentric is placed concentric with the axle, to altogether stop the motion of the valve, and, therefore, that of the engine also. In No. 2, Fig. 66, the eccentric is supposed to be moved across the axle in a direction not at right angles to that of the radius of the crank. This would allow for a certain amount of lap and lead in, say, forward gear, but the valve, it will be seen, would be correspondingly late in back gear, and it would, indeed, be thus impracticable, with considerable lap and lead, to work the engine backwards. In No. 3, Fig. 66, the eccentric is supposed to be capable of movement across the axle, but not in a line diametrically across. The line *a e*, in which the eccentric is made to move, when required, across the axle, is still at right angles to the radius of the crank, as in No. 1; but it is parallel to the corresponding diametrical line *a e*, No. 1, and is removed from it by a distance equal to the sum of the lap and lead of the valve. In No. 3 the centre of the eccentric may be made to occupy any position in the line *a e*, and thus, whether in forward or back gear, the given lap and lead of the valve will be always preserved. The positions of the eccentric will be better seen, and the corresponding movement of the valve understood, from Fig. 67, although any number of successive positions may be given to the eccentric. Here, then, we have the means of expansive working, viz., variable travel combined with lap and a constant lead upon the valve. Referring back to Fig. 65, the mode of adjusting the eccentric in different positions with reference to the centre of the axle will be understood. A double wedge is made to slide along the axle, and, in its motion, to force the eccentric (through which it works) into the successive positions required. The sliding wedges are held in any desired position by suitable latches or fastenings on the reversing lever. Dodds' wedge-motion eccentric, adjusted as in Fig. 67, has been adopted as a variable expansion gear in a considerable number of modern locomotives—among others, in some very heavy tank engines designed by the late Mr. Brunel for the Vale of Neath Railway. In his specification, Dodds referred to variable expansive working; and this he proposed to effect, not by applying his wedge motion, properly adjusted, to the main valve, but by employing a separate eccentric adjustable in a line reaching diametrically across the axle, and working a separate expansion valve. The practical qualities of Dodds' wedge-motion eccentric have been found to be reasonably satisfactory, and it is not improbable that it may come into extensive use in future practice. It takes up very little room under the boiler, has few joints, is cheaply fitted, and answers all the geometrical conditions of valve movement fulfilled by the link motion. It has been objected to in some cases as liable to jam between the wedges and eccentric, and as difficult to reverse with steam on. With well-fitted surfaces, proper attention to oiling, and with properly proportioned reversing apparatus, these objections would be obviated.

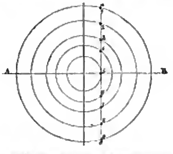

Fig. 67.—Successive Positions of Dodds' Eccentric for Lap and Lead.

At this stage of the history of variable expansion gear it will be as well to regard the effect of variable travel of the valve in conjunction with lap and lead. In order to effect the expansion of steam within the cylinder, both induction ports should be closed; although a certain expansive effect, due to "wire-drawing" the steam, results when one induction port is only open to such an extent that the entrance of full-pressure steam into the cylinder is less rapid than the enlargement of space due to the movement of the piston. In the consideration of expansive working, it is more convenient, however, to refer the commencement of expansion to the actual closing of the induction port through which the steam to be expanded has been admitted. With a valve having no lap, there is but one point in its range of motion at which the steam is excluded from both induction ports. Except for a moment, therefore, in each complete stroke or throw of the valve it will be admitting steam to either one or the other end of the cylinder; and in this case there can be no expansion, except by "wire-drawing" the steam, and this result could be and was effected by Dodds' variable gear of 1839. When, however, lap is added to the opposite edges of the valve there must be a period, in each stroke or throw of the valve, during which steam can enter neither end of the cylinder, and this period will be greater or less in a certain relation to the proportion which the sum of the lap, at both ends of the valve, bears to the range of its movement.

The valve shown in Fig. 68 can effect no expansion of the steam, except where lead is given, the effect of which we need not yet consider. In Fig. 69 the valve has a lap at each end equal to the width of the induction ports, and it is evident that while the valve is moving from one extreme of its range of movement to the other, say in the direction of the arrow, no steam will be admitted to either end of the cylinder from the time the edge *a* of the valve has closed the port 2, until the opposite edge *b* has opened the port 1. During this time the piston will be moving from the right-hand end of the cylinder, or that in communication with port 2, towards the end in communication with port 1; and during this time the steam already acting upon the piston will be expanding. If the lap at each end of the valve bear but a small proportion to its whole range of movement, the period during which the steam is expanding will bear but a moderate proportion to that occupied in making the full stroke of the piston. If, however, the total movement of the valve be but little greater than the sum of the lap at both its ends, then the period during which the steam is expanded in the cylinder at each stroke will bear a large proportion to the whole time of making the stroke. Let us first suppose that the total movement of the valve is 4 inches, the lap at each end being 1 inch. The eccentric must be so set as to have the edge *b* of the valve, Fig. 69, in line with the outer edge of the port 2, at the beginning of a stroke of the piston from the right to the left-hand end of the cylinder. This position of the eccentric may be found as shown in Fig. 70, which represents the crank A, the position and circular path of the centre of the eccentric, and the corresponding positions of the crank and eccentric when the edge *b* of the valve, Fig. 69, again closes over the port 2. It being supposed that the valve is connected directly with the eccentric, the motion to be given by the latter, and which motion is commonly called the throw, should be 4 inches. If the valve had no lap the centre of the eccentric would be at *i*, the radius of the eccentric, *b i*, being at right angles to the crank A *b*. But with 1 inch lap on the valve the centre of the eccentric must be 1 inch to the left of *i*, with the piston at the right-hand end of the cylinder, and the crank pin at A; and in this position the eccentric will have the valve ready to admit steam at the beginning of the stroke. The angle A *b e*, between the crank and the eccentric, is now fixed, and in the present case it happens to be 120 degrees. As the piston moves from the right hand of the cylinder, Fig. 69, towards the left-hand end, the crank and eccentric will turn in the direction of the arrow, and for a time the eccentric will move the valve, Fig. 69, so as to open the port 2 wider and wider, until when, the eccentric having gone from *e* to *n*, it will open the valve no further. The valve will then begin to close the port 2, Fig. 69, and when the centre of the eccentric has reached e^1, it will have placed the valve where it will have just closed the port 2. The position of the valve at this instant will be the same as when the centre of the eccentric was at *e*, but the valve will be moving in the opposite direction. And when the centre of the eccentric is at e^1, the crank, preserving its fixed angle with it—an angle in the present case of 120 degrees—will be in the line A^1 *b*. If we disregard the inconsiderable effect produced by the angularity of the connecting rod, we may find the position of the piston, corresponding to the crank at A^1, by raising an ordinate from A^1 to *m*, this ordinate being at right angles to A B, representing the stroke of the piston. As in the present case the crank will have moved through 120 degrees from A to A^1, the corresponding distance, A *m*, moved by the piston will be found to be exactly three-fourths of its whole stroke, represented by A B. Thus, with 4 inches travel of the valve and 1 inch lap, and without lead, the steam is cut off at three-fourths of the stroke. Now, we can suppose the amount of expansion to be capable of being varied by either of two modes. By increasing the lap of the valve and the angle, A *b e*, of the eccentric with the crank, the steam would be cut off still earlier on the stroke, and that portion of the stroke during which expansion would be going on would be correspondingly increased. But no means of varying the lap while the engine is at work are known. The effect which would be produced by varying the lap may, however, be obtained by altering the travel of the valve, and this may be readily effected while the engine is working, either by Dodds' wedge-motion eccentric or by the link motion. Fig. 67 illustrates the successive positions of the eccentric in Dodds' motion. If we place the eccentric where it will give a travel of 2·83 inches to the valve, the lap still remaining 1 inch, the relative position of the crank and eccentric will be as in Fig. 71. The centre, *e*, of the eccentric will, in effect, be advanced more around the axle, although it will have been moved only on the line

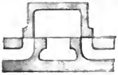

Fig. 68.—Slide Valve, no lap.

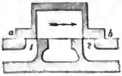

Fig. 69.—Slide Valve, lap equal to width of induction ports.

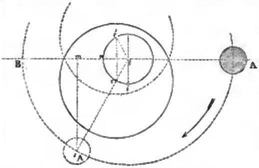

Fig. 70.—Diagram for setting Eccentric for Lap Valve.

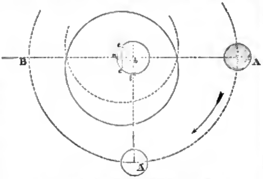

Fig. 71.—Diagram for setting Eccentric for Lap Valve.

e e^1 of Fig. 70. The angle A *b e* will now be 135 degrees instead of 120 degrees. The eccentric, when it reaches e^1, will have moved the valve, Fig. 69, so that the port 2 will be again closed; and it will be found that when the centre of the eccentric is at e^1, Fig. 71, the crank will be at A^1, and (disregarding the obliquity of the connecting rod) the piston will be at half stroke. In the same way the effect of different amounts of travel on the valve, with constant lap, may be found, the point of cutting off being earlier in the stroke, and the consequent expansion greater as the travel of the valve is shortened. The action of the link, in giving a varying motion to the valve, is, in its effect, the same as that of Dodds' wedge-motion eccentric.

Where lead is also given to the valve, or, in other words, where the steam ports are already opened to some extent, when the piston is about to begin a new stroke, the position of the eccentric requires to be still further in advance of the crank. If it be desired to have the steam ports ¼ inch open when the piston is just ready to begin a new stroke, the centre, *e*, of the eccentric in Figs. 70 and 71 must be ¼ inch further from *i* than it would be without lead. By ¼ inch further is to be understood that the line *e* e^1, in which the centre of the eccentric is supposed to be found, must be ¼ inch further from the line *i b*. In this case the steam is cut off still earlier on the stroke. The whole effect of these adjustments of the eccentrics and proportions of the valve will be more fully considered in another chapter.

In 1841 the late Mr. Robert Stephenson patented an arrangement of the machinery of locomotive engines, in which he introduced the valve gear shown in Fig. 72. A top and a bottom gab were fixed to the valve spindle, and the ends of the eccentric rods were held apart by a straight link, as shown. There was no novelty in the principle of this gear; indeed, its action was substantially the same as that of Roberts', Fig. 63, although the former was much simpler than the latter. But Stephenson's reversing gear of 1841 was a visible suggestion of the link motion of 1843. The straight distance link in Fig. 72 necessarily received the very bodily motion which a curved link would have had if connected to the same eccentric rods. The pin in the end of the backing eccentric rod could never, when engaged with the lower gab on the valve spindle, exactly follow the rectilinear motion of the latter, as the eccentric rod end swung in the curve described by the weigh link. Hence there was a constant chafing in the lower gab when the engine was in back gear. If, instead of the gabs, a slotted swivelling block had been fixed to the end of the valve spindle, and if the distance link had been curved so as to have worked freely through this block in reversing, the whole arrangement, regarded merely as a reversing gear, would have been much improved. But in the improved form would have been contained the best type of the link motion, the solid curved arm or bar (proposed by Mr. Roberts in 1832), applied as in Messrs. Humphrys and Tennent's marine engines. The peculiar value of a variable travel of the valve, notwithstanding what Gray had done in 1838, had not become known, however, in 1841.

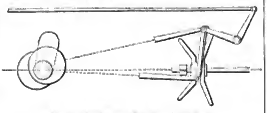

Fig. 72.—Stephenson's Reversing Gear, 1841.

In 1842 or 1843 Mr. Williams, of Newcastle, is said to have designed the attempt at a link motion, shown in Fig. 73, although this is copied from an *ex parte* drawing sent to the Glasgow *Practical Mechanic and Engineer's Magazine* in 1846, by Mr. William Howe, of Newcastle, who claimed the form of link motion now in general use as his own invention. An apology is necessary for introducing such a mechanical blunder into these pages; but it has been said, by those who ought to know better, that Williams' contrivance "contained the germ of the link motion, and needed only to be cleverly worked out to render it the most felicitous acquisition which the locomotive has received since the combination of the multitubular boiler and blast pipe"! It does not, on the contrary, require a second glance to perceive that the notable "design" reproduced in Fig. 73 did not permit of making half a revolution of the driving wheels except by tearing this pseudo "link motion" to pieces. The assurance originally given by Mr. Howe, that it had never been carried into practice, was quite unnecessary. Even could this form of "link" have held together while at work, it does not appear that it was schemed with any reference to expansive working. The eccentrics occupy a relative position to each other in the sketch, which appears to show that neither lap nor lead was contemplated, while also the slot, intended for the movable block of the valve spindle, is straight, instead of being curved to the radius of the jointed valve rod, as it should have been if intended to work at any point intermediate between the two ends.

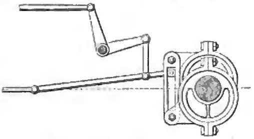

Fig. 73.—Williams' Link Motion (?), 1842.

On the 15th February, 1842, Mr. Crampton patented a form of variable expansion gear, which, while producing the same movement of the valve as Gray's, was much simpler, and, indeed, even simpler than the link motion in its present form. Mr. Crampton proposed the single fixed eccentric and the general form of the Carmichael reversing gear employed by Goldsworthy Gurney in 1826; but Mr. Crampton formed curved slots in the ends of Gurney's link, and thus worked his lap valve with any required throw, the lead varying at the same time as in Gray's gear.

We at last come to the link motion as first applied to

Messrs. Robert Stephenson and Co.'s engines in 1843, and shown in Fig. 74. Although this link possessed no theoretical, but possibly some practical, advantages over Dodds' wedge eccentric motion, as now adjusted for lap and lead, it is to be remembered that, down at least to 1843, the most valuable feature of Dodds' wedge gear remained undiscovered. It had never then been applied for variable expansive working, otherwise than by "wire-drawing" the steam. The link, therefore (and although James' claims are not to be overlooked, its general introduction is due to the Messrs. Stephenson alone), was, if not the most original, at least the most admirable form of variable expansion and reversing gear known in the history of the Steam Engine. No other form of valve gear ever introduced so many, and, at first sight, such complex, but at the same time valuable, conditions of valve movement. No innovation, in respect of locomotive engines, ever divided practical and professional opinion so completely; and none has at last so firmly established itself in general favour. It is difficult to believe that the link motion can ever be superseded in its general principle, although there are improved forms of the link, such as Allan's, shown in Plate III., which may be preferred to the original of twenty years ago. It is only, indeed, through the force of habit that we continue to employ the slotted link instead of the solid curved bar, which is the simplest and most mechanical form which this gear can have, unless a solid straight bar with Allan's arrangement of working parts be preferable.

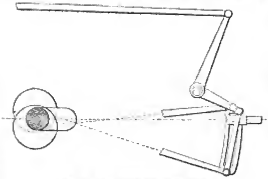
Fig. 74.—Link Motion (by Howe), 1843

It has already been observed that the link motion was, in all probability, a chance discovery at Newcastle in 1843, as it was eleven years before in New York. That this was the case may indeed be shrewdly inferred from the fact that it was never patented, notwithstanding that the late Mr. George Stephenson, Mr. Robert Stephenson, and Mr. Howe were, in other cases, ever ready to secure to themselves the utmost lawful protection for the fruits of their ingenuity. Their names frequently appear in the records of the Patent Office, yet, it must be said, rarely in connection with inventions of substantial value. Bearing in mind that George Stephenson and William Howe (notwithstanding that Dodds had patented a three-cylinder locomotive in 1839) were associated in a patent for an improved three-cylinder engine in 1846, it may be fairly supposed that, had they comprehended the link in 1843, they would not have failed to patent that also. The link motion was, we may be certain, first designed as a reversing gear only, and as a neater job than the straight distance piece and gabs patented by Mr. Robert Stephenson two years before. It is easy to understand that the former was employed as a structural improvement upon the latter plan, and that its value as an expansion gear was discovered by an engine-driver who chanced to work it, or by some "young gentleman" in the drawing office, who had occasion to develop its various positions upon a broad surface of "double elephant" or "antiquarian" paper. Had a combination of such priceless qualities been thoroughly known when, or before, it was first adopted, we may be sure that, if it were not patented, it would at least have formed the subject of a "Paper" at the Institution of Civil Engineers. The link motion was not "born," however; it "growed," and it is understood to have been in public use for a time without any knowledge of its singular and admirable properties.

The general effect of the motion imparted to the valve, in the different positions of the link, is like that to be obtained from Dodds' wedge gear already illustrated; and indeed the action of a valve worked by a link may be studied, where great exactness is not required, with the help of Fig. 67. But although the results are nearly the same, the respective means of producing the motion of the valve differ in principle quite as much as in form. In Dodds' gear the valve is always worked by one and the same eccentric; but with the link, the valve, except when in full gear, partakes of the motions of two eccentrics, the bodily motion of the link being the same as that of the straight distance link shown in Fig. 72. This bodily motion of the link will be understood in the following manner:—Without reference to *varying* the travel of the valve, it is necessary (as it would also be with any other form of reversing gear, say that of Roberts, Fig. 63) to fix the position of the eccentrics upon the axle, so that, with a given amount of lap, they shall always have the induction port open, or ready to open, at the beginning of the stroke of the piston. Without either lap or lead, the radii of the forward and backing eccentrics would be exactly opposite to each other on the axle, and both would be at right angles to the crank, as in Fig. 75, where the line of the crank is shown by A B, and the respective eccentries are marked *Fd* and *Bk*. If a link were to be employed with eccentries thus set, the mid length of the link would have no motion beyond that resulting from the crossing of the eccentric rods, when the eccentrics shown in Fig. 75 made half a revolution. This effect is shown in four successive positions of such a

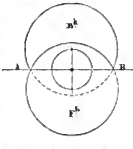

Fig. 75.—Setting of Eccentrics: no lead nor lap.

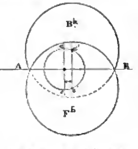

Fig. 76.—Setting of Eccentrics: lead and lap.

link, in Fig. 77. Were the eccentric rods infinitely long, and the radii of the eccentrics tolerably short, the mid length *a* of the link would thus have no motion. As, with rods of the usual length and eccentrics of ordinary throw, this effect is comparatively slight, the lap and lead are to be provided for in the setting of the eccentrics, as shown in Fig. 76. Here the line A B represents the direction of the crank, the crank pin being supposed to be in the direction A. The smaller circle shows the path, concentric with the axle, described by the centres of the eccentrics, *a* and *e* being the respective positions of the back and front eccentrics without lap or lead on the valves. Their corresponding positions with lap and lead are to be found in a line parallel to *a e*, and removed from it (in this case to the right hand) by a distance which, if we neglect the effect due to the crossing of the eccentric rods, will be equal to the sum of the lap and lead at each end of the valve. Thus, with 1-inch lap and ⅛th-inch lead, the line $a^1 e^1$ will be 1⅛ inches removed from, and parallel to, *a e*, and at the points where the line $a^1 e^1$ intersects the circular path described by the radii of the eccentrics will be the positions of the centres of the back and forward eccentrics respectively, when the crank is lying towards A, in the direction A B. These diagrams suppose that the crank, lying towards A (and in which direction also the cylinder is supposed to be placed), turns under, as in going forward; and it is also assumed that the motion of the eccentric is communicated directly to the valve, instead of being reversed by intervening rocking levers. If a rocking shaft be used, so as to reverse the motion of the eccentrics, the line $a^1 e^1$, Fig. 76, will be to the left of *a e*, and the centre of the forward eccentric will be above, and that of the backing eccentric below the line A B. In other words, the centre of each eccentric will be diametrically opposite to the position shown in Fig. 76.

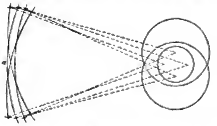
Fig. 77.—Link Motion: no lap, the lead being due to crossing of Eccentric Rods; or with equivalent lap and no lead.

With the eccentrics set as in Fig. 76, it will be readily seen how the valve, acted upon by the link, will always, except when in full gear forward or back, partake of the motions of both eccentrics. Fig. 78 represents the successive motions assumed by the centre line of a link in working a valve with lap and lead. The upper end of the link is worked from the forward eccentric, and when the end of the valve spindle is placed in gear, with any part of the link intermediate between its two ends, the valve will derive a motion compounded from that of the two eccentrics, but which motion does not in reality differ from that due to the corresponding position of the single eccentric in Fig. 67. When the valve spindle is placed in gear with the middle of the link, Fig. 78, the consequent motion of the valve will be the same as that due to the position marked 5 in the line *a e*, Fig. 67; and the other positions of the end of the valve rod between the top and bottom of the link, Fig. 78, nearly correspond in effect with the respective positions of the eccentric, between those marked 1 and 9, Fig. 67. The loops described by the ends of the link are caused by the rise and fall, in a curve, of the end of the hanger which supports it. The adjustment should be such as to make this irregularity as little as possible.

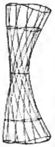
Fig. 78.—Successive Motions of Link.

The link motion, of the form shown in Fig. 74, has the peculiarity also of varying the lead of the valve, according to the successive positions in which the link is placed with relation to the end of the valve spindle. This effect may be understood from Fig. 79, representing the link in full and mid gear respectively, at opposite half revolutions of the eccentrics. When the centres of the forward and backing eccentrics are at a^1 and b^1 respectively, the link in full gear occupies the position shown by the line C D. In mid gear it rises to A B; the top of the link, while rising into mid gear, describing the curve C A, with the radius a^1 C. The difference, whereby the curve C D falls short of the curve A B at C, represents a like increase of lead in changing from full gear to mid gear. When the eccentrics, after half a revolution, acquire the positions *a*, *e*, the corresponding position of the link is shown by A^1 B^1 in mid gear, and by *c d* in full gear, and the distance by which *c* falls short of A^1 B^1 represents a like variation of lead on the valve at the beginning of the return stroke. In the diagram, with the forward eccentric working the top of the link, the lead will increase from full gear to mid gear, the increase being greater as the throw of the eccentrics and length of the link are increased, and less as the eccentric arms are lengthened. If, however, the backing eccentric work the top of the link, the lead will diminish from full gear to mid gear, and for a geometrical reason which at once appears when the link is dropped from A to C, or from A^1 to *c*, with the upper eccentric rod moving from the centre a^1 instead of e^1, or from *a* instead of *e*. The increase of lead in approaching mid gear (the forward eccentric being sup-

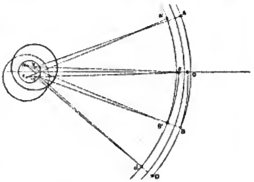

Fig. 79.—Variation of Lead in Full and Mid Gear.

posed to work the top of the link) is commonly reckoned an advantage, as, under ordinary circumstances, the greatest amount of lead is thus given when the engine is going at its highest rate of speed.

Another effect produced by the link motion is, not only to close the communication with the exhaust port, before the completion of the stroke of the piston (as is the case with all lap valves, however worked), but to close it earlier in the stroke, as the travel given to the valve is lessened. And when the communication between one end of the cylinder and the exhaust port is closed earlier in the stroke, that with the opposite end of the cylinder is also opened correspondingly earlier. The reason of this will appear when it is considered that every change in the extent of the travel given to the valve corresponds, in effect, to a change in the position of the eccentric with reference to that of the crank. Here, again, the action of the link may be compared with that of Dodds' wedge motion, the link in mid gear giving the same motions as the single movable eccentric when in the position marked 5 in Fig. 67; while full gear and the intermediate degrees of movement agree with the other positions, from 1 to 9, in the same diagram. Where the valve has no inside lap (and no "clearance"), both ends of the cylinder are closed to the exhaust port when the valve is at the middle of its travel, and when, therefore, the effective radius of the single eccentric, Fig. 67, would be at right angles to the centre line of the engine. But the effective radii of the eccentric in different positions make different angles with the crank, as would be seen by drawiog these radii from the centre of the axle successively to the various points numbered 1 to 9 (although any other number of points may be taken) in Fig. 67. By bringing these radii successively into a position at right angles to the centre line of the engine, it would be seen that, with the shorter radii, the crank would be further from the positions corresponding to the ends of the stroke of the piston than when the full throw of the eccentric was employed in working the valve.

The foregoing explanations of the general action of the link may, to some extent, anticipate the substance of a further chapter, to be devoted to the geometrical illustration of valve gear; but the discovery of the several properties of the link, and their demonstration, have formed important incidents in its history subsequent to 1843. The action of locomotive valve gear had not then been studied with the aid of the indicator, and it may be safely asserted that many engineers were content to employ the link motion long after its first appearance at Newcastle, without any accurate knowledge of its effects when worked otherwise than in full gear. There were many, too, who, having worked out the effect of the movement given to the valve in each notch of the reversing gear, were inclined to condemn the link on the ground of its wire-drawing the steam, and of its closing the exhaust too early in the stroke on one side of the piston, and opening it too soon on the other. So, too, the slow movement of the valve, in intermediate gear between full gear and mid gear, was a ground of objection, as, indeed, it still is. A practical objection, too, was that the face of the cylinder, over which the valve worked, became hollowed, because of the different ranges of movement given in different positions of the link. In many cases it was thought necessary to provide a hardened face for the cylinder, this face being formed upon a plate of steel fastened to the cylinder, and having openings through it answering to the steam and exhaust ports.

Experience has shown that most of these objections to the link are really unimportant. When working the valve so as to cut off very early in the stroke, the link may no doubt be charged with a certain loss of expansive effect due to wire-drawing the steam; but when it is considered that the full boiler pressure may be, and, indeed, generally is, admitted upon the piston at the beginning of its stroke, and this, too, even with the least travel of the valve, it is clear that the expansive effect may be very good after all. Nor at high speeds does the early opening of the exhaust port, the link being supposed in nearly mid gear, appear to be attended with any considerable loss, inasmuch as the steam is generally already expanded to a point not greatly above that of the constant back pressure upon the piston. In some cases, indeed, the steam is found to have expanded below the pressure of the atmosphere before opening the communication to the exhaust port; the expansion being thereby attended with a positive loss. With the ordinary proportions of valve gear, the steam at the higher speeds of the engine is never completely exhausted from the piston, notwithstanding that the opening to the exhaust port may be wide open before the piston reaches the end of the stroke. As long as the steam is escaping from the cylinder during the next or return stroke, the back pressure, whatever it may be, is a complete loss; but as soon as the exhaust port is closed on the return stroke, any steam then remaining is compressed by the piston; and although the available pressure, or the effect derived from the engine, is thereby diminished, the exhaust steam so compressed is saved for the next stroke. The only loss resulting from what is technically called "compression in the cylinder," is due to the circumstance that the steam is not ordinarily worked so as to obtain, from that which has been compressed, as much power as was expended in compressing it. The loss, however, due to compression is not great, although it has been very commonly assumed that the amount of compression is the measure of a total loss of steam. This compression, furthermore, brings the piston quietly to rest, preparatory to the beginning of the succeeding stroke, and thus, in conjunction with lead (to the effect of which that of compression is analogous), it relieves the suddenness of the strains which would otherwise come upon the piston and other working parts, while it also tends to counteract the effect of their momentum, which, at high speeds, and but for counterweights, would produce great unsteadiness of the engine upon the rails. The early closing of the communication with the exhaust port, more especially when the link is working nearly in mid gear, is by no means therefore a ground of serious objection to the link motion, as was for a long time contended by many who would either not employ it at all, or, if so, under protest.

The link motion requires great nicety in the adjustment of the weigh shaft and lifting rod—a nicety which is often neglected, and in the absence of which the steam is admitted and exhausted at very different points in the forward and back strokes respectively. The adjustment of the various parts of the valve gear will be fully treated of in a subsequent chapter.

The link motion was rapidly introduced into English locomotive practice, wherein it is now employed nearly to the exclusion of all other forms of valve gear. The Continental engineers, however, and the Americans, hold their ground stubbornly, and for a long time, upon the various objections which have been discussed on the previous page; and it was not until 1849 that the link was first adopted by a leading American maker; and not until 1855 that its application became general on the other side of the Atlantic. The French and Germans were nearly as late in their recognition of the advantages of the new gear.

In the period between 1843 and 1847, no striking changes were made in the locomotive engine; and it may even be said that, with the link motion, the engines of twenty years ago were not greatly inferior in design and structure to those of the present day. The Messrs. Stephenson, it may be noted in passing, made about this time a considerable number of "long-boiler engines," of the kind described in their patent of 1841. Their 15-inch cylinder engines had 150 tubes 1⅝ inches in diameter and 13 feet 6 inches long, all the wheels being placed between the fire box and smoke box. It is not now certain with what view this class of engines was designed, but it has been often stated that the long boiler was intended to secure the presumed advantage of the greatest practicable extension of the tube surface. The actual extent obtained, 861½ square feet, was certainly large at the time for a 15-inch cylinder engine, and it would be reckoned liberal now, notwithstanding that a greater allowance is sometimes given, as for example, 1,236 square feet for the 16-inch cylinder engine shown in Plate III. Where, however, a great area of tube surface is given now, it is usual to provide a corresponding extent of heating surface in the fire box; but this was not the case in the Stephenson engine of 1841. There is evidence, moreover, in much of the contemporary writing of that time, that tube surface was then reckoned a fair measure of evaporative power, and that therefore the greater the extent of tube surface, the greater the evaporation. It would be unfair to suppose that the assumption had been embraced that, with a given consumption of coal or coke, the rate of evaporation might be indefinitely increased by extending the tube surface. But at the same time it must be borne in mind that even twenty years ago engineers had a most imperfect knowledge of the real heating value of coke and coal—a value beyond which nothing can be gained by any amount of ingenuity in boiler making, or by any extension of heating surface. There were, now and then, official reports upon engines and boilers, in which an alleged evaporation of 16 lbs. of water per pound of coal was stated as a fact which might be implicitly depended upon! It was not until 1842 that the American Government appointed Professor Walter R. Johnson to examine and report upon the heating value and other qualities of the coals employed for the United States' vessels of war, and it was not until 1846 that, with this example, Sir Henry De la Beche and Dr. Playfair were entrusted with a like commission in the interest of the Royal navy. There need be no doubt that, at the time the long-boiler engine was designed, the ultimate heating value of fuel was believed to be much greater than it really is; and it was natural to suppose (the moderately low temperature of the escaping products of combustion in the smoke box and in the chimney being no doubt overlooked) that from a further increase of tube surface would result a like gain in evaporation from the same weight of fuel. It was the fate of the multitubular boiler, as of most successful inventions, to be greatly over-rated, and it was then natural to regard tubes and heat as almost equivalent terms. Mr. Stephenson, it is true, had made an experiment as early as 1830,[*] in order to ascertain the relative value of the heating surface of the fire box and tubes respectively, but this experiment did not show the effect which would attend an addition to the tube surface alone. An ordinary locomotive boiler had its top removed, and the water surrounding the fire box was separated by a tube plate from that which surrounded the tubes. A fire being raised, the quantity of water which disappeared by evaporation from the respective compartments of the boiler in a given time was noted, and it appeared that each square foot of fire box surface was equivalent in evaporative effect to 3 square feet of tube surface. Even with this evidence, it appears to have been assumed that for every 3 square feet of tubular heating surface added to the boiler an additional weight of water would be evaporated equal to that boiled away upon 1 square foot of the fire box surface. In 1842 the late Mr. Dewrance, of the Liverpool and Manchester Railway, made a further and more suggestive experiment upon the relative value of tubular and and fire box heating surface. He divided a small boiler into seven different compartments, one being the fire box portion, the second a length of 6 inches of the tubes, while the other compartments were formed by tubes plates separating successive lengths of 1 foot of the tubes, which were 5 feet 6 inches in their total length. In a letter from Mr. Dewrance to Mr. C. Wye Williams (dated February 1, 1858, and published in the appendix to *Three Reports on the Use of the Steam Coal of the Hartley District*) it appears that the first compartment of 6 inches of the tubes was about equally efficient, for a given surface, as the fire box; the second compartment along the tubes was but about one-third as effective; while, says Mr. Dewrance, "in the remaining four compartments the evaporation was so small as to raise a doubt on my mind whether it had any value at all. In fact, I came to the conclusion that the first 6 inches of the tubular series had more evaporative effect than the remaining 60 inches." Mr. Williams made a nearly identical experiment in his laboratory, with a tube 5 feet long and 3 inches in diameter, passing through a vessel divided into five compartments, each 1 foot in length. A powerful fire was raised, and the temperature of the waste heat, on leaving the tube, was as high as 800°, yet the boiling point was never once reached in the compartment furthest from the fire. Water of 50° temperature having been supplied to the several compartments at the beginning of the experiment, the quantities respectively evaporated from them in a period of three hours were as follows:—No. 1 (next the fire), 117 oz.;

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[*] *Vide* Wood *On Railroads*, third edition, p. 403.

No. 2, 92 oz.; No. 3, 73 oz.; No. 4, 64 oz.; and No. 5 (furthest from the fire), 63 oz. In other experiments, where the temperature of the waste heat escaping from the end of the tube was as low as from 475° to 515°, the rate of evaporation in the compartment furthest from the fire was relatively very much less than in the case just instanced.

The long-boiler engines did not therefore attain to any remarkable economy of fuel, while it must have required a strong blast to overcome the friction of the air through such small and so long tubes. Few are willing to give to the world the records of their unsuccessful experiments, and we must be satisfied, therefore, in the absence of precise details in this case, to believe what has been generally reported of these engines, to wit,—that they could only keep themselves in steam when worked with blast pipes much smaller than usual, and consequently with a considerable back pressure upon the pistons. They had another fault, moreover,—that of great overhanging weight at the ends, the fire box and foot-plate being entirely behind the trailing axle, while there was the usual proportion of weight forward of the leading axle. The late General Pasley, who, at the request of the Government, made very close observations upon the subject of railway speeds, in 1845, found that the long-boiler engines could not run steadily at speeds greater than 45 miles an hour, and in his opinion they were unsafe beyond that speed.* Some of the long-boiler engines had inside cylinders, and four coupled 5-feet wheels, besides a pair of leading wheels; while others had horizontal outside cylinders, with a single pair of driving wheels near the fire box, and two pairs of smaller wheels forward, the cylinders being placed at some distance behind the smoke box. It became necessary, at last, to place a pair of trailing wheels beneath the foot-plate. The locomotive practice of the Messrs. Stephenson was less settled than that of the other leading makers. It may be that the firm just named were entitled to credit for their various, and certainly well-meant, efforts to improve the locomotive engine, but none of the features which characterized their designs have survived, with the single exception of the link motion. Messrs. Stephenson successively produced engines with long boilers and with short boilers, with inside and with outside cylinders, with inside frames and with outside frames, and with nearly every known variety of working gear.

The engines made by Messrs. Sharp, Roberts, and Co., and by their successors, Sharp, Brothers, and Co., more nearly represented what for many years was the favourite pattern upon English railways. Fig. 80 is from a drawing published in 1847, and a large number of engines of this class are still at work. Although a growth from Mr. Roberts' designs, it bears evidence, in many points, to the taste and careful attention to detail of Mr. Beyer, who for some time after Mr. Roberts' retirement was the principal engineering authority at the Atlas Works. To Mr. Beyer is commonly ascribed the deep, solid inside framing now preferred, and having the axle guards welded on, instead of being secured by bolts or rivets, as was at one time practised. Mr. Roberts had designed a deep framing, each girder of which was a single plate, as early as 1833; and a like framing is shown in the drawings attached to Mr. Stephenson's specification of 1841. But, until the steam hammer had come into use, it was not reckoned an easy task, nor would it have made a good job, to weld on the axle guards. In Fig. 80 the position of the dome, near the chimney, gives the engine a hump-backed appearance, not in keeping with the otherwise symmetrical and dainty features of the design. There was at one time much difference of opinion as to the best place for the dome, it having been found that when the steam was taken from directly over the fire box, as in most of the earlier engines, a good deal of water was carried over into the cylinders. This result was no doubt

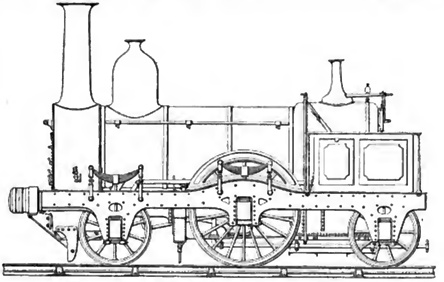

Fig. 80.—Sharp, Brothers, and Co.'s Engine, 1847.

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* Vide *Proceedings, Institution of Civil Engineers*, vol. viii., 1849—Discussion on Mr. Crampton's Paper.

due, in a great measure, to the fact that the steam room, and the area of the water level from which the steam rose, were insufficient; so much so, that near the point where the steam was drawn off, the water (when the engine was working) was always in a state of violent agitation. It is related of Bury's engines, that, on opening the regulator on starting, the apparent level of the water in the boiler, as shown by the gauge glass, often rose as much as 10 inches, thus showing a violent foaming of the water in the boiler, consequent upon the efforts of the steam to disengage itself. It was attempted to avoid the consequences, rather than to remove the cause, of this internal commotion, and hence the dome was placed further forward by some makers, until, as in Sharp's engines, it was set nearly against the chimney. Dr. Church, in 1837, provided two domes for his engine, the "Eclipse," and he appears to have taken off his steam in part from each dome. Messrs. Hawthorn, as early as 1839, obtained a patent for the use of a perforated steam pipe, intended to draw the steam from all parts of the steam room alike, and thus to dispense with the dome altogether. Mr. Gough, in his steam carriage patent of 1828, proposed the perforated pipe; but the Messrs. Hawthorn were probably the first to use it, as they certainly were to point out its effect. In their specification, filed May 15, 1840 (patent dated November 21, 1839), they stated that they perforated the steam pipe with holes ⅛th inch in diameter, the holes being furthest apart next to the smoke-box end of the steam pipe, and increasing in number towards the fire box. The collective area of all the holes was to be about one-fourth more than the whole cross-sectional area of the steam pipe, and this proportion, the patentees observed, had already been found by them to give the best results in practice. The perforated steam pipe was long employed in Messrs. Hawthorn's practice, and it is occasionally resorted to now, as in Messrs. Beyer, Peacock, and Co.'s engine, Plates III. and IV.; but it appears to be the more general conclusion that priming, so far from being cured, is rather aggravated by the perforated steam pipe, although very much depends upon the means adopted to throw down the water which tends to rise to it.

In the same specification Messrs. Hawthorn described steam-drying and superheating apparatus for locomotives. They formed a steam chamber for this purpose in the upper part of the smoke box, extending this chamber partly around the smoke box, and in some cases hot-air tubes were to pass through the steam-drying chamber. Messrs. Hawthorn at that time made some of their engines with return tubes in the boiler, the chimney being placed over the fire box, while the exhaust steam pipe was led from the smoke box directly through the boiler in order to reach the chimney. This plan of keeping the exhaust pipe out of sight, as well as of keeping it hot at the expense of the steam in the boiler, has long been in favour among makers of portable agricultural engines. One of Messrs. Hawthorn's sections shows seventy tubes from the fire box to the smoke box, and forty-five tubes of greater length returning from the smoke box, and passing over and past the sides of the fire box into a second and smaller smoke box, made partly over and partly behind the fire box. The roof of the inside fire box was arched to the form of a semicircle in cross section, and no stay bars were applied. In another drawing, two narrow fire boxes, with semicircular crowns, were shown with a water space or mid-feather between them. This arrangement was afterwards introduced into the large engines made at Wolverton for the London and North-Western Railway. Messrs. Hawthorn pointed out the advantages of the arched crowns; but their only object in proposing double fire boxes appears to have been to "lessen the intensity of the heat," and to obtain increased heating surface. The chief value of the double fire box is when it is employed in conjunction with a combustion chamber, and when the fires are fed alternately, as in the South-Eastern passenger engine, by Mr. Cudworth, Plate XI. The combustion chamber, as subsequently employed in the London and North-Western engines, was patented, June 2, 1846, by Stubbs and Grylls, of Llanelly, although their patent was not probably respected in its use.

During the period between 1840 and 1846, over which the foregoing record extends, many complicated arrangements of valve gear were patented and used, more especially on the Continent. It has not been thought necessary to reproduce them here; but for those who wish to examine them, it may be said that a large number are figured and described on pages 108 and 165 of the *Bulletin de la Société d'Encouragement* for 1846, as also in the *Practical Mechanic and Engineer's Magazine* for the same year, (vol. v.) A large number of these were elaborations of Allen's expansion valve, Fig. 58, page 53 of the present work. As the name of M. Meyer appears in connection with more than one of these, it may be said

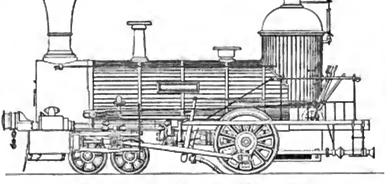

Fig. 81.—Meyer's Engine for the Continental Railways, 1848.

that he was, and still is, an ingenious adaptator of the American practice. Fig. 81 represents one of his locomotives, from drawings in the *Bulletin de la Société d' Encouragement* for 1849 (repeated also in *Der Ingenieur* for the same year), and which is characterized by many of the features which William Norris had introduced into his short practice at Vienna. In Meyer's engine the centre pin or pivot of the truck or bogie was placed near the leading axle; the stroke of the pump plunger could be varied by means of a jointed lever worked from a curved arm, as in Roberts' variable expansion gear; and the smoke box was diminished to a small size by a plate or partition across it just above the top row of tubes, the chimney extending downward to this partition.

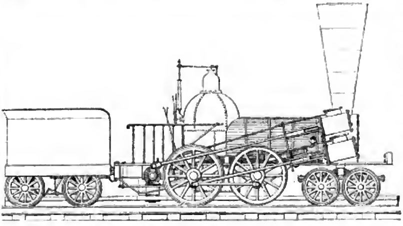

Fig. 82.—G. E. Sellers' Locomotive for "Middle Rail," 1847.

Although it was customary in 1845-50, as it is yet, to ridicule Blenkinsop's useful rack rail, where great traction at slow speed was required, there was here and there one who foresaw, what is now every day becoming more apparent, that some mode of increasing the hold of the engine upon the rails was desirable, in order to enable locomotives to be worked over very steep gradients—gradients which in some situations were inevitable, and which engineers were more ready to adopt than in the early period of railway history. As long ago as 1830, Mr. Charles Vignoles and Captain John Ericsson were associated together in a plan for a middle rail up steep inclines, and for a pair of wheels on vertical axes, gripping this rail on each side, and which, by their forcible revolution, would carry up a train where the ordinary driving wheels of an engine would slip without effect. This plan is figured and described in Hebert's *Encyclopædia*. A like plan was embodied in the arrangement patented by George Escol Sellers (in the name of A. V. Newton), July 13, 1847, and of which an elevation is given in Fig 82. Here the engine had the usual pair of cylinders, working coupled driving wheels on the level and moderately inclined portions of the line; while, by means of an extra pair of cylinders, and a corresponding pair of wheels working on vertical axes, and gripping a middle rail on steep inclines, the effective adhesion might be increased to almost any extent. In Fig. 82 the mode of working the gripping wheels from the upper pair of cylinders and through bevel gearing will be clearly seen. The middle rail shows about 4 inches above the ordinary rails, and behind the spokes of the wheels, and one of the gripping wheels is seen in elevation just below the stout bearing which separates it from the horizontal bevel wheel. Each pair of cylinders had its own regulator or "throttle-valve," the separate handles for which are shown. Mr. Sellers constructed four or five engines upon this plan for the Panama Railroad, possibly before the present moderate gradients of that line had been determined upon. The engines were never brought into use there, and were subsequently altered or broken up. More recently two more engines upon nearly the same plan were made for the railway of a coal company in Pennsylvania; but these, for some reason (probably the bankruptcy of the purchasers), were never set to work. In January, 1863, Mr. J. B. Fell, of Sparkbridge, Lancashire, patented a locomotive much upon the general plan of Sellers', but having horizontal cylinders, and two pairs of coupled gripping wheels, driven direct without bevel gearing, the connecting rods to the gripping wheels working in a horizontal plane. An engine upon Mr. Fell's plan was made at the Canada Works, Birkenhead, where the details were worked out by Mr. Alexander, now of the Millwall Iron Works. This engine was worked experimentally on the Cromford and High Peak Railway, where it ascended inclines of 1 in 13, and with a considerable load. The engine was intended for a line of unusually steep inclination, which it was proposed to make over Mont Cenis, on the railway from Lyons to Turin. A difficulty with this, as with Sellers' engine, is in the irregularity of the power given out from the ordinary driving wheels and the gripping wheels respectively. When working together, one or the other manifests an almost constant tendency to slip, unless the admission of steam to the respective pairs of cylinders is regulated with unusual nicety.

In 1847 began a violent controversy, which, although originating in engineering questions chiefly, soon came to be waged in parliamentary committees, and between laymen and in the newspapers. The broad and narrow gauges had at last met, rail to rail, and the great companies representing the respective systems were determined to contest the prize of railway extension. If the alleged advantages of the 7-feet gauge could have been established, the power of the Great-Western Company would have been enormously increased. The very chance of this made the issue a most important one to the London and North-Western Company, and no means were neglected, therefore, for making the most of the arguments on both sides. At one time the gauges of the railways in the United Kingdom were of very various widths. That of the Ballochney line, in Scotland, was 4 feet 6 inches; the Liverpool and London, and most of the lines laid out by the Messrs. Stephenson, were 4 feet 8½ inches between the rails; the London and Blackwall, Eastern Counties, and Northern and Eastern lines were 5 feet wide; the Dundee and Arbroath and Arbroath and Forfar lines were 5 feet 6 inches wide; the Ulster Railway had a gauge of 6 feet 2 inches; and that of the Great Western was 7 feet. The Irish gauge has since been fixed at 5 feet 3 inches. The real contest of 1847-48 lay, however, between the 4-feet 8½-inch and 7-feet gauges. It was sought to show that the resistances were less upon the broad than upon the narrow gauge; and although it might therefore have been supposed that lighter engines would be required on the former, it was urged that the increased width permitted of increasing the size and weight of the engine to an extent unattainable on the narrow gauge. The inside-cylinder arrangement was generally preferred between fifteen and twenty years ago, or before it had become customary to properly counterweight outside-cylinder engines; and it was, no doubt, more convenient to lay out a very large inside-cylinder engine on the broad than on the narrow gauge. Mr. Brunel had adopted 18-inch cylinders, 24-inch stroke, and 8-feet driving wheels for the Great-Western express engines, while nothing upon this scale had, down to 1847, been attempted upon the narrow gauge; and, indeed, such engines had not been required. It became an important object, however, while the gauge contest raged, to prove that as heavy trains might be taken at as high a rate of speed upon the narrow as upon the broad gauge, and that, indeed, as large engines were admissible upon the former as upon the latter.

Mr. F. Trevithick, son of the famous Cornish engineer, was among the first to grapple practically with the supposed difficulties in reconciling a large boiler and a high driving wheel with a gauge of only 4 feet 8½ inches. His engine, the "Cornwall," designed in 1847, and shown in the Great Exhibition of 1851, is represented in section in Fig. 83, from drawings kindly furnished at the author's request. The "Cornwall" was built for the London and North-Western line, on which it is still working. It has 17½-inch cylinders, 24-inch stroke, and a pair of 8-feet 6-inch driving wheels. The boiler has been replaced, however, by one over the driving axle. The prominent feature of Mr. Trevithick's design has been often brought forward as original, since 1847, by those who may, or who may not, have known of the existence of the "Cornwall."

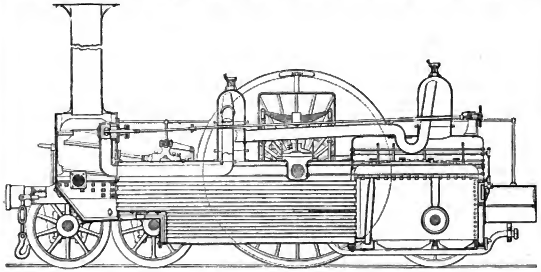

Fig. 83.—Trevithick's Engine, 1847.

In 1848, Mr. Crampton (who had patented the prominent feature of the "Cornwall," in 1842, and who had obtained several patents for bold and novel combinations of locomotive engines in 1845, 1846, and 1847) altered one of the long-boiler engines on the South-Eastern Railway, by placing the driving wheels behind instead of in front of the fire box. This single alteration was all that was required to complete what is now known as a "Crampton engine," the outside cylinders being already laid horizontally at some distance behind the smoke box. But Mr. Crampton worked out his favourite idea with much boldness, and with a respectable amount of success.

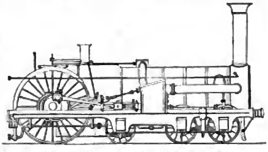

Fig. 84.—Crampton's Engine, "Liverpool," 1848.

Late in 1848 one of his engines, made by Bury, Curtis, and Kennedy, commenced working upon the London and

North-Western Railway. This engine, the "Liverpool," shown in Fig. 84, was undoubtedly the largest ever constructed for passenger traffic, and it settled the question of the admissibility of the largest engines upon the narrow gauge. It had 18-inch cylinders, 24-inch stroke, 8-feet driving wheels, a boiler containing 300 tubes 2$\frac{3}{16}$ inches in diameter and 12 feet 3 inches long, while the total heating surface was 2,260 square feet. The extreme wheel base was 18 feet, and the total weight, in working order, 35 tons. The fire grate, it may be noted, was 21½ square feet in extent, and the heating surface in the fire box was 154 square feet. The "Liverpool" was reckoned equal to a load of 180 tons at 50 miles an hour, but its great weight and length told seriously upon the permanent way, which, in 1848, was not fish-jointed as now, while steel rails were unknown. The "Liverpool" class of engine did not, therefore, meet with favour in England, but it was taken up by the French engineers, and the American makers returned, in some instances, to their former practice of placing a single pair of driving wheels behind the fire box. With the exception of the small short-boiler engines shown in Fig. 48, page 46, the first engines made in the States to resemble the "Liverpool" were not designed until after the construction of that class of engine had become well known. This is mentioned here, in order to settle a disputed question of dates. Messrs. M. W. Baldwin and Co. made an engine, the "Governor Paine," upon this general plan, with 17-inch cylinders, 20-inch stroke, and a pair of 6½-feet driving wheels, which was delivered to the Vermont Central Railroad in August, 1849. The same makers subsequently sent three "Crampton engines," of smaller dimensions, to the Pennsylvania Central Railroad (September, 1849). Norris Brothers made seven engines for the Camden and Amboy Railroad, each with a single pair of 8-feet driving wheels, and a six-wheeled truck. The first of these, with 13-inch cylinders and a 34-inch stroke, was sent from the makers' shops, April 17th, 1849. The next of the class had 13-inch cylinders and a 38-inch stroke, and was started in December, 1849. The last of the series, delivered in April, 1853, had 14-inch cylinders and a 38-inch stroke. The 13-inch cylinder engines of this construction weighed, empty, 40,754 lbs.; and, loaded, 49,253 lbs. Of the weight loaded, 18,469 lbs., or about 8¼ tons, were on the driving wheels, with 30,784 lbs., or say 13¾ tons, on the truck, making 22 tons in all. These engines had boilers only 36 inches in diameter, and which were formed of plates but $\frac{5}{16}$ inch thick, the seams of rivets being made to run spirally around the boiler, as in those made by Mr. Wright, of Goscote. The fire box was very long, for the purpose of burning anthracite coal, and the roof of the outer fire box sloped backward and downward, so as to pass under the driving axle.

The gauge contest, as is well known, was settled, in its political aspects at least, in favour of the narrow way. Upon the 7-feet gauge, and notwithstanding the claim made for it in respect of light resistance to traction, the engines have for many years been of great size, without, however, accomplishing proportionately heavier work. The Great-Western express engines of the class in use since 1848 are shown in Fig. 85. They have 18-inch cylinders, 24-inch stroke, 8-feet driving wheels; and those of the largest size have 21 square feet of fire grate, 153 square feet of fire box surface, and 305 tubes 2 inches in diameter and 11 feet 3 inches long, presenting 1,799 square feet of external surface, or a total heating surface of 1,952 square feet. The barrel of the boiler is 4 feet 6 inches in diameter; but from the width of the gauge the cranks are so far apart upon the axle that the bottom of the boiler is but about 4 feet 8 inches above the rails. These engines present no unusual features of design, other than in their great size. It has been said of them that they would evaporate from 300 to 360 cubic feet of water per hour; and they are known to have worked up to quite 1,000 indicated horse-power; but this maximum effort was sustained for but a short time, and the alleged hourly rate of evaporation is probably overstated. At 60 miles an hour, 1,000 indicated horse-power would correspond to a mean effective pressure of 77·16 lbs. per square inch upon the pistons of the engine shown in Fig. 85.

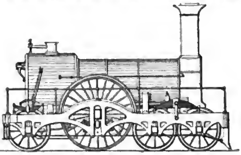

Fig. 85.—Great Western Express Engine, 1851.

Even as late as 1851 it was contended that, the inside-cylinder engine being then preferred, the narrow gauge could not, in respect of the admissibility of this favourite type, after all compete in power with the Great-Western line. No railway authority appeared willing to multiply engines of the "Liverpool" class, which was taken to represent rather what was possible than expedient. The London and North-Western Company was not long, however, in bringing into use the very largest inside-cylinder engines of which the narrow gauge admitted. These, of which the first commenced work in 1852, were on six wheels, and weighed 34¾ tons in working order. The cylinders were 18 inches in diameter, the pistons having a stroke of 24 inches. The driving wheels were 7 feet, and in subsequent engines 7 feet 6 inches in diameter, and were loaded to 14 tons 6 cwt. The weight on the leading wheels was 11 tons 18 cwt., and on the trailing wheels 8½ tons. The boiler was 51 inches in diameter, and contained, in the earlier engines, 305 tubes 1¾ inches in diameter and 7 feet long; and in the later engines (one of which was exhibited in the International Exhibition of 1862), 214 tubes 1⅞ inches in diameter and 9 feet 4 inches long. The fire box was, however, the remarkable feature of the boiler, it being of great size and in two compartments, with arched crowns, as in Hawthorn's patent of 1839. A combustion chamber extended also for a considerable distance into the barrel of the boiler. In the earlier engines no less than 260 square feet of surface were thus obtained in the fire box and combustion chamber, while with longer tubes and a shorter combustion chamber, 242 square feet were secured. The grate was in two parts, each 7 feet long, and of a collective area of 26 square feet. The fire boxes were 6 feet 3 inches below the crown plates, and the centre of the boiler was 7 feet 5½ inches above the rails, or at nearly the same level as the top of the driving wheel. It was only by recessing the combustion chamber, to permit the cranks to work under it, that the boiler was kept to this height. These engines were intended to burn coal; but although they did so with but little smoke, the rate of evaporation was found, in 1854, to be but 5·83 lbs. of water per pound of coal burnt, while the temperature of the waste heat in the smoke box sometimes reached 1100°. In further experiments made by the author, in 1857, upon one of these engines, the steam, which was at 115 lbs. on starting, fell in half an hour to 80 lbs., and finally to 68 lbs., from which it was some time in rising again to 80 lbs. The train was one of 17 carriages, or say 100 tons, and the speed, between London and Bletchley, only 30 miles an hour, including two stops. The engines of the class under notice originally had superheating apparatus in the smoke box, and the pistons were of wrought iron, forged solid with the rod, as in an engine made by Mr. Hackworth in 1850. In the earlier engines india rubber springs were applied in conjunction with outside bearings, but subsequently inside framing and steel springs were adopted.

These great weights upon the narrow gauge were not reached at a jump—at least in the inside-cylinder engines. Those made for the Birmingham and Shrewsbury line in 1848, had (for 15-inch cylinders, 20-inch stroke, and 5-feet 7-inch wheels) boilers 4 feet in diameter, containing 172 tubes 1¾ inches in diameter and 11 feet 6 inches long, while the fire box measured 4 feet 2½ inches by 3 feet 6 inches on the grate, and was 4 feet 7 inches deep. Timothy and J. W. Hackworth's engine, the "Sanspareil," made in 1849-50, had 221 tubes in its boiler, most of which were 2 inches in diameter; and the "Bloomer" class of engines, made for the London and North-Western line in 1850, had 16-inch cylinders, 22-inch stroke, 7-feet wheels, and weighed 28¾ tons, of which 12 tons 7 cwt. were on the driving, 9 tons 18 cwt. on the leading, and 6½ tons on the trailing axles. The fire box had 142 square feet of heating surface, and the boiler was 50 inches in diameter, and contained 195 tubes 12 feet long, and presenting 1,152 square feet of surface.

In 1853 a number of large tank engines were made for the Bristol and Exeter Railway (of 7-feet gauge), and of dimensions and upon plans which entitle them to historical record. They are shown in Fig. 86. They had 16½-inch cylinders, 24-inch stroke, driving wheels 9 feet in diameter, and two trucks of 4 wheels each, the truck wheels being 4 feet in diameter. The boiler was 4 feet ½ inch in diameter, and contained 180 tubes 1$\frac{11}{16}$ inches in diameter, the barrel of the boiler being 10 feet 9 inches long. The driving wheels had no flanges. India rubber springs were applied under all the axle bearings. The weight of the engine, loaded, was 42 tons, or rather less than the aggregate weight of an ordinary engine and tender of the same power and capacity. These engines were made by Rothwell and Co., of Bolton-le-Moors; and Mr. Pearson, who designed them, informs the author that he has eight of them now running, with the original wheels of 9 feet diameter still under them. Mr. Pearson has reported that they have run at the rate of 80 miles an hour with complete steadiness, and that for a mileage of upwards of 80,000 miles the first engine of the series averaged a consumption of only 21¾ lbs. of coke per train mile.

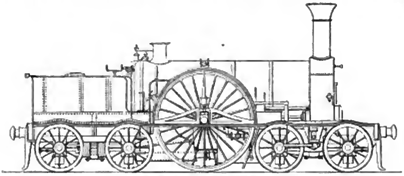
Fig. 86.—Bristol and Exeter Engine, 1853.

About this time the author, in view of the preference for large driving wheels, designed an express engine of unusual size and construction, and of which a brief description may not be out of place here. It appeared to him that the objections to all previous plans for very large-wheeled engines upon the narrow gauge were these:—If the boiler is over the axle, the engine is top-heavy; if the boiler is beneath the axle of any pair of driving wheels less than 12 feet in diameter, the form of the fire box and disposal of the tubes are unsatisfactory; while, if the driving axle is behind the fire box, hardly enough weight can be secured for adhesion. A large wheel, moreover, implies a large boiler, and not only is it difficult to provide room in a narrow-gauge engine for a very large boiler, but, beyond a certain size, there is a chance of failure under high pressures. The author, therefore, proposed the arrangement shown in Fig. 87. The two boilers, each 43 inches in diameter, are collectively of the same capacity as one of the same length and 5 feet in diameter, while, other things being equal, the strength of the former would be nearly one-half greater than that due to the latter. The heating surface is ample for all wants, the weight is well over the axle, the centre of gravity is reasonably low, the driving wheels can be easily removed when required,

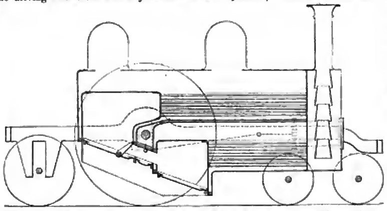

Fig. 87.—Design for Express Engine (1854); 18-inch cylinders, 30-inch stroke, 10 feet driving wheels, weight 36 tons.

and the working parts can be conveniently arranged. Both fire boxes can be fired as one, and by a simple arrangement all the fuel can be kept well clear of the short intermediate length of dead plate between the fire boxes. By means of an apparatus known as a "petticoat pipe" an equable draught would be maintained through the tubes of both boilers. Each boiler would have its own water level and steam space, the steam from the lower boiler being led by a large pipe into the dome above. Room would be obtained for long bearings on the driving axle, and for large springs above them. The valve gear would be external to the wheels. The four wheels in front would be combined in a swivelling truck. This design was thrown out as a general solution of an alleged difficulty, and not in view of any particular case.

The most striking example of the employment of large driving wheels on the narrow gauge is that furnished by the engine "L'Aigle," designed by MM. Blavier and Larpent, and made by Messrs. Gouin, of Paris, for the Western Railway of France. This engine was exhibited in the Paris *Exposition* of 1855. Fig. 88 is sketched from a large tracing, for which the author is indebted to his friend M. Flachat, late engineer-in-chief to the Chemin de Fer de l'Ouest. This engine had 16½-inch cylinders, 31½-inch stroke, and four coupled wheels of the great diameter of 9 feet 4 inches (2·85 mètres). The form of the boiler appears in Fig. 89, and it was such as to permit of the convenient disposal of the axles; but as the engine never answered any useful purpose, probability is given to the natural suspicion that such a boiler could not produce steam rapidly and economically. Cylinders of the same diameter, moreover, with a 22-inch stroke and 6½-feet driving wheels, would have given the same tractive force with the same mean pressure of steam, and in this case, too, the velocity of the pistons would now have been the same at any given speed of the engine. Thus, at 60 miles an hour, the pistons of "L'Aigle" would now move at the rate of 945 feet per minute, which would have been their velocity also with a 22-inch stroke and

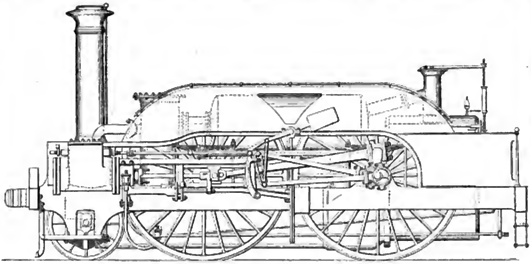

Fig. 88.—MM. Blavier and Larpent's Engine "L'Aigle," 1855.

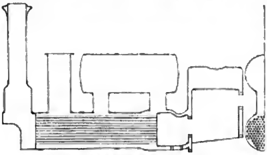

Fig. 89.—Boiler of Engine "L'Aigle."

6½-feet wheels. Again, it is doubtful whether any advantage was gained by coupling wheels of such great size, and which supported so large a proportion of the weight of the engine. With the high average effective pressure of 100 lbs. per square inch upon the pistons, the tractive force of this engine would have been 7,657 lbs.; and, taking the adhesion of the wheels as only one-sixth of the weight, this would require 20½ tons upon the coupled drivers. But as a mean pressure of 100 lbs. per square inch upon the pistons would not often be attained, and as, with the occasional use of sand on the rails, an adhesion of even one-fourth of the weight upon the driving wheels may be had, it would have been better to have employed but a single pair, loaded, say, to 13 tons. The engine shown in Plate VII. is found to work well upon the heavy gradients of the Caledonian Railway, and its principal dimensions may be compared therefore with those of "L'Aigle." The Caledonian engine has 17¼-inch cylinders, 24-inch stroke, and driving wheels 8 feet 2 inches in diameter, loaded to 14 tons 11 cwt. when in working order. The tractive force, with a mean effective pressure of 100 lbs. per square inch upon the pistons, would be 7,286 lbs., equal to quite $\frac{2}{9}$ths of the whole weight for adhesion. So, too, with an ordinary pattern of express engines, having 16-inch cylinders, 20-inch stroke, and 6-feet driving wheels, a weight of 12 tons upon the latter is always reckoned ample. Yet, with 100 lbs. mean effective pressure per square inch of the pistons, the tractive force would be 7,111 lbs., equal to more than $\frac{1}{4}$ths of the adhesion weight. These comparisons will show that the striking dimensions of "L'Aigle" were needless.

It need hardly be said that engines of the dimensions and weight of the "Cornwall," "Liverpool," the Wolverton pattern, "L'Aigle," &c., were exceptions to the common practice of 1847-55. Yet, whether from the influence of these examples, or, which is more probable, the growing needs of railway traffic, passenger engines of upwards of 30 tons weight are now not uncommon; and notwithstanding that the usual pressure in locomotive boilers has been increased from 80 lbs., in 1847, to 120 lbs., and even to 150 lbs. in some cases, express engines of from 28 tons to 30 tons weight are now preferred upon all the great lines. When, however, a decided movement commenced, in 1848, in favour of heavier engines, counter claims were advanced in support of an opposite practice. Mr. William Bridges Adams had made, for the engineer of the Eastern Counties Railway, a light engine to be used in journeys of inspection over the line, and, although the whole weight of the engine was at first but 23 cwt., it attained a speed, upon a level, of 41 miles an hour; and, when taken to the Birmingham and Gloucester line, ascended the Lickey incline of 1 in 37. This engine, shown in Fig. 90, burned but from 2¾ lbs. to 3½ lbs. of coke per mile when carrying seven persons.

The total length of this engine was 12 feet 6 inches; the frame was hung below the axles, the floor being only 9 inches from the rails. The four wheels were 3 feet 4 inches in diameter, and the two cylinders were 3½ inches in diameter, with a 6-inch stroke of piston acting upon a cranked axle.* The upright cylindrical boiler was 19 inches in diameter and 4 feet 3 inches high. The fire box was 16 inches in diameter and 14 inches high, and there were thirty-five tubes 1½ inches in diameter and 3 feet 3 inches long, the heating surface in the fire box being 5½ square feet, and in the tubes 38 square feet. The "Express," although it moved itself only, with its coke, 40 gallons of water, and seven passengers, was considered so successful that Mr. Adams received orders from the Eastern Counties, Bristol and Exeter, Cork and Bandon, and other railway companies, for light engines, having a carriage on the same frame, although, in a later arrangement, the engine worked a peculiar system of

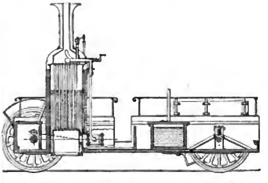

Fig. 90.—Adams' Locomotive "Express" for Eastern Counties Railway, 1847.

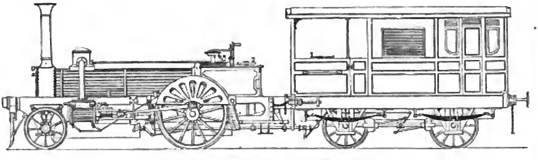

Fig. 91.—Adams' Light Engine and Carriage Stock, 1850.

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* All of the steam machinery of this and several of Mr. Adams' subsequent engines was designed by Mr. Edward Reynolds, Mr. Adams' pupil at the time, and who is now engineer to Messrs. Naylor, Vickers, and Co.'s extensive steel works at Sheffield.

composite carriages, one of which served as tender, brake, and luggage van, and first and second class compartments. This arrangement is shown in Fig. 91, the peculiarity being more apparent in the carriage than in the engine. The carriage, too, was of the great width of 9 feet, and was planned to accommodate a large number of passengers in proportion to the dead weight carried. The engine shown in Fig. 91 (made for the Londonderry and Enniskillen line) had 9-inch cylinders, 15-inch stroke, 5-feet driving wheels, and 3-feet leading wheels, the wheel-base being 10 feet. The boiler was 2 feet 6 inches in diameter, and contained eighty-six tubes 1¾ inches in diameter and 10 feet 10 inches long, the fire box having 37½ square feet of heating surface, with 5¼ square feet of grate. It was worked at what in 1850 was considered the very high pressure of 120 lbs. per square inch. A tank under the engine held 250 gallons of water, and another under the brake contained 600 gallons. The engine weighed 9 tons. The performance of the steam carriages on the Eastern Counties and Bristol and Exeter lines was such as to show the great commercial advantage of a diminution of dead weight where practicable, but dynamically the results attained were not remarkable. The Eastern Counties steam carriage "Enfield," which commenced working in January, 1849, was made to accommodate eighty-four passengers; yet the wheel-base was but 20 feet long, and the weight, with the engine in working order, but 15 tons 7 cwt., which would be increased to about 21 tons by the weight of the passengers and luggage,—engine, coke, water, and passengers being upon the same frame. Mr. Adams had constructed several carriages 40 feet long and 9 feet wide, which, supported upon a novel arrangement of eight wheels, seated no less than 116 passengers. One of these carriages, as well as a brake van, was attached to the "Enfield," and, thus accommodating 150 passengers, or more when required, it worked for a considerable length of time, the maximum speed being 37 miles an hour. When working without the extra carriage, the average consumption of coke, including that burnt in raising steam and while standing, was 11¼ lbs. per mile; while the actual consumption when running was but about 6 lbs. per mile. Other engines, commonly employed in a like service, then burnt from 20 lbs. to 35 lbs. per mile. The saving was due almost entirely to the suppression of the tender, and to the lightness of the machine, considered both as an engine and a carriage. The later examples of Mr. Adams' practice, illustrated in Fig. 91, were distinguished from the then prevailing standards by compactness and elegance of design, although a resemblance might be traced to Messrs. Stephenson's outside-cylinder long-boiler engine, with the small middle wheels removed. There was merit, under the circumstances, in this removal; and thus it was, no doubt, that a "design" of Messrs. Stephenson's, published in a well-known work in 1855, was almost identical with the engine shown in Fig. 91, notwithstanding that 11-inch cylinders and 18-inch stroke were substituted for 9-inch and 15-inch respectively.

While there is an important commercial advantage in the diminution of the relative dead weight of rolling stock of whatever kind, it is not the less a fact that, regarded as an engine only, the light locomotive works at a decided disadvantage, and that, in proportion to the effective horse-power exerted, the larger sizes, if they have equally good design, structure, and management, work at a less cost for fuel, repairs, and attendance than the light engines of which so much was said in 1849. The steam gig shown in Fig. 90 never worked, probably, with an average pressure upon the pistons greater than 50 lbs. per square inch, except at the slowest rates of motion. With this mean effective pressure, at a speed so great even as 30 miles an hour, less than 7½ indicated horse-power would have been exerted; but when it is considered that this estimate involves that of 92 lbs. as the total resistance to the motion of a gross weight of but about 1½ tons, it is doubtful if as much as 7 horse-power was exerted. The consumption of coke would not be less than 10 lbs. per indicated horse-power per hour. In the case of the "Enfield," even when drawing its extra carriage and brake van, and with a full complement of 150 passengers, the gross weight would fall short of 40 tons. (The carriage seating 116 passengers weighed but 7¼ tons.) Whether we make the high allowance of 75 lbs. per square inch as the mean effective pressure upon the pistons, or whether a mean resistance of 25 lbs. per ton moved at 35 miles an hour be taken, the horse-power exerted upon the pistons would be but 87; but as the consumption of coke was but 280 lbs. per hour in this service, this would show a very fair economy of fuel for the power exerted. If we disregard the friction of the working parts of the "Enfield," the utmost progressive force or draught which it could exert at the peripheries of the driving wheels, and with the high allowance of 75 lbs. mean effective pressure per square inch upon the pistons, would be but 960 lbs.; whereas a 14-inch cylinder engine, with 20-inch stroke and 5-feet wheels, would, under the same circumstances, exert a tractive force of 4,900 lbs., or more than five times as much as the "Enfield." But as the 14-inch cylinder engine would have but about three times the heating surface of the "Enfield," its horse-power may be taken as three times as great, and this, too, would be about the relative proportion of the weights of the respective engines. In no case could the light engine possess any dynamical advantage over its rival, except when the latter was worked to less than its proper power. The argument in favour of engines weighing less than say 12 tons was, indeed, that those of greater weight were not worked up to their full power. In many cases they are not; but railway experience upon the largest scale has conclusively shown the vital importance of such a command of power as will insure that an engine keeps its time under all the circumstances of a capricious traffic, and in all, or nearly all, weathers. Now, by a return for a single week in May, 1849, the greatest number of passengers at any one time in a main line train on the Eastern Counties Railway was 231, while the least number was but 7. For the branch lines the numbers were 82 and 3 respectively. A like irregularity of traffic exists to this day on most railways; but the engine must, as a rule, be sufficiently powerful to take the heaviest train, excepting, of course, in cases where an unusual concourse of passengers can be foreseen and provided for. With the ordinary construction of carriages, 231 passengers would represent a train of nearly or quite 70 tons weight, and to take this, at a mean rate of 35 miles an hour, over a line of average gradients and curvature would require a tractive force of at least 2,500 lbs., in addition to the resistances of the engine and tender. A tractive force of 2,500 lbs. at 3,000 feet per minute (or rather more than 34 miles an hour) corresponds to 227 horse-power. No engine weighing less than 23 tons, and having less than 750 square feet of heating surface, could be depended upon for so large a disposable power beyond that required for moving itself and tender. The ordinary experience, moreover, upon main line railways proves that, instead of 70 tons, passenger trains of a maximum weight of 100 and even 120 tons must often be taken, and the tractive force required to move them increases with the speed; and at a mean rate of 40 miles an hour, and upon very moderate gradients, upwards of 400 horse-power may be expended upon the train alone. Here an engine and tender of from 45 to 50 tons weight are indispensable, in order to exert the requisite 500 to 600 indicated horse-power. It may be possible to effect a saving of dead weight in proportion to the number of passengers carried; but, while this does not come strictly within the province of the locomotive engineer, it may be as well to remember that the tendency is constantly towards increased space and accommodation, and that the English public, at least, dislike the gregarious style of conveyance represented by great floor-area and many-seated compartments in railway carriages. The success to which Mr. Adams' light engine and carriage stock attained in 1850 was due in a very great measure to the lightness of the carriages, and consequently diminished resistance to be overcome, and to the then unusual boiler pressure (120 lbs. per square inch) at which the light engines were worked. There were many, however, who hastily assumed, as the conclusion to be drawn from Mr. Adams' practice, that a 10-ton engine could perform as much work as one of twice the weight; and in America very light engines were again constructed, to be attached to ordinary passenger trains. One of these had indeed been made by Messrs. Baldwin and Co. for the Vermont Central Railroad as early as January, 1848. This engine, the "Adams," had 5½-inch cylinders, 14-inch stroke, 4-feet driving wheels, and a truck, and weighed only 5 tons. The fire box was 18 inches long, 18 inches wide, and 2 feet deep, and the boiler, 22 inches in diameter, contained 45 tubes 6 feet long and 1⅛ inches in diameter. Mr. Griggs, of the Boston and Providence Railroad, made two tank engines (which commenced work early in 1851) with 9-inch cylinders, 16-inch stroke, 4½-feet wheels, 4·55 square feet of fire grate, 29·3 square feet of fire box surface, and 113 tubes 8 feet long and 1½ inches in diameter, presenting 357·7 square feet. These engines weighed 11 tons 6 cwt. 1 qr. 10 lbs. in working trim; and when at rest had 3 tons 3 cwt. 3 qrs. 10 lbs. on the single pair of driving wheels. The pulling bar was so connected, however, to levers pressing upon the boxes of the driving axle as to draw a considerable additional weight upon the driving wheels when the engine was at work, and the adhesion was thus increased in a ratio following the increased resistance to be overcome. One of these engines, at a public contest between ten locomotives at Lowell, 1st and 2d October, 1851, drew two long eight-wheel passenger carriages, containing eighty-one passengers, over 9 miles of nearly level line at a mean speed of 42 miles an hour, the maximum speed maintained for 3 consecutive miles being at the rate of 48 miles an hour. The weight of the carriages and passengers drawn was 18 tons.* The Cumberland Valley Railroad was afterwards worked for some years by a class of very light engines, and with a success for which even the favourable gradients and light trains of that line hardly accounted, but which may be better understood when it is observed that the fastest trains now run (in June, 1864) average but 17 miles an hour, including stops, the "accommodation" trains being worked at a still slower rate. The subject of light engines may be dismissed for the present, with the remark that the question of engine-weight is practically one of engine-power; that heavy engines exert rather more power in proportion to their weight than light engines; and that, under all circumstances, the total resistance of the train is the exact measure of the necessary power to be exerted by the engine beyond that required for moving itself and tender. Something should be added to the remark that a heavy engine exerts more power per ton of its own weight than a light engine. Were the heavy engine merely an enlarged copy, upon a definite scale, of a light locomotive, its weight would be as the cube of the increase of size, while the heating surface, which practically is the measure of locomotive power, would increase only as the square. Thus, if every dimension of a 10-ton engine were doubled, its weight would necessarily be increased to 80 tons, while its grate area, heating surface, and blast orifice, and therefore its power, would be increased only four-fold. But in practice the increase of heating surface is more rapid than the increase of weight. It has been seen how 2,000 square feet of heating surface have been provided in a 35-ton engine; but no locomotive engineer would attempt or expect to arrange 1,000 square feet of fire box and tube surface in a locomotive weighing only 17 tons 10 cwt. With the exception of a few bold strides in the direction of great locomotive power, the weight of locomotive engines has been gradually and even timidly increased to meet the imperative necessities of increased weight and speed of railway trains. In this matter locomotive engineers, instead of leading, have notoriously been led.

Tank engines will, no doubt, some day claim a page in history. The earliest steam carriages, however, carried their own water and fuel, and the first tramway and railway engines would have been equally independent of a "tender" but for the weakness of the trams and rails originally laid down, and over which it was therefore necessary to distribute the weight as much as possible. The "Novelty," of 1829, was a tank engine; and so was Dr. Church's engine, the "Eclipse," of 1837. An impression exists, however, that tank engines are of modern origin, and they have been widely attributed to Mr. W. Bridges Adams. He employed them, to some extent, where the whole weight did not exceed 10 tons, but he publicly recorded his objections against their use for heavy trains. Tank engines came into extended use between 1847 and 1850 for branch traffic, and for service about stations, and there is reason for believing that they will be much more generally employed in future practice than they have yet been. If the wheels of an engine are already loaded to the limit of the economical supporting power of the rails, it would be imprudent to add more weight to them in the form of water tanks and coke boxes; but engineers are not now unwilling to place 12 tons, and even 14 tons, upon a single pair of driving wheels, and upon at least one great metropolitan line cast-iron dead weight is placed under the foot-plate, purposely to increase the adhesion. If, then, the weight of the necessary supply of fuel and water can be added to that of the engine without exceeding the safe, or rather prudent, limit of load upon a single axle, it is better to do so than to provide a separate tender. A mere tank, to contain 1,200 gallons of water, need not weigh more than 1 ton; and coal boxes, to contain even 1 ton of coal (a supply for from 60 to 100 miles), would hardly exceed half a ton. For the mere purpose of carrying water and coal not more than 1½ tons, or, making the very largest allowance, 2 tons, need be added to the weight of the engine; yet an empty tender of the capacity assumed would weigh at least 8 tons. 1,200 gallons of water and 1 ton of coal would amount together to a little more than 6¼ tons; and 8 tons, therefore, may be taken as the weight to be added to a 22-ton engine in order to dispense with a loaded tender weighing between 14 tons and 15 tons. It is true, the above estimate makes no allowance for the increased strength of some of the parts of the engine necessary to support the maximum additional load of 8 tons; but 1 ton at the most thus added to the weight of the parts of the engine is known to answer all the purposes of 8 tons in the form of the tender. There is thus a clear saving of 7 tons in the case assumed, which fairly represents the ordinary conditions of practice. But, on the other hand, if the engine is intended for a given service, the very saving of 7 tons is a saving of at least one-tenth of the weight of an ordinary main line passenger train; and in any case a saving, at high speed and over ordinary gradients and curves, of at least 30 indicated horse-power. If advantage be taken of this to employ, as may be done, an engine of 1½ tons to 2 tons less weight than would otherwise be required, there is a further saving of, in some cases, 8 or 10 indicated horse-power by the diminished resistance of the engine itself. The allowance of 1,200 gallons of water is, moreover, large for a passenger tank engine, and half that quantity would in many cases be considered ample. The objection to tank engines, apart from the additional weight brought upon wheels already heavily loaded, is that, from the constantly varying weight of the fuel and water, the adhesion weight of the engine is often disadvantageously affected,—that perhaps 6 tons or 8 tons of fuel and water, representing so much adhesion weight, may have been consumed when adhesion is most required. By a skilful distribution of the tank weight, so that it shall not be wholly supported upon the driving wheels, this objection is to some extent overcome.

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* *Official Report of the First Exhibition of the Middlesex Mechanic Association*, Lowell, 1852.

Stephenson and Howe's three-cylinder engine was patented in 1846, and one or two were subsequently made for one of the northern lines. Fig. 92 is a section through the smoke box and cylinders of one of these. It appears that this arrangement was not adopted with the once common idea that the alternate action of the *steam* upon the cylinder covers, in outside-cylinder engines, caused sinuous motion; nor was it perceived that the then usual unsteadiness of this class of engines was due to the *momentum* of the piston, piston rod, cross-head, and connecting rod, alternately acting in opposite directions on opposite sides of the engine. It was formerly believed that outside-cylinder engines were necessarily unsteady, and that the evil was not to be cured by any system of counterweighting. Mr. Fernihough had practically shown the effect of counterweighting the reciprocating as well as the revolving parts as early as 1845; and by 1849, at least, the soundness of his views was generally admitted by the leading members of the profession, as is evidenced in the discussion, early in that year, on Mr. Crampton's paper, at the Institution of Civil Engineers. But there were many who held to the fallacy that the very action of the steam, alternately admitted upon the covers of widely spread cylinders, caused a right and left motion, it being overlooked that an equal pressure was transferred through the piston to the axle boxes, there to act in an opposite direction, and with precisely the same leverage, upon the mass of the engine. As for the momentum of the reciprocating parts, a few pounds of counterweight would have served a better purpose than the extra cylinder and working parts in the arrangement shown in Fig. 92, and which, according to Stephenson and Howe's specification, were adopted to counteract the alternate lifting of the opposite sides of the engine, causing a rocking motion, and due to the upward thrust upon the guide bars. The plan was not perpetuated, although Norris, in America, adopted it for two or three engines made for the Philadelphia, Wilmington, and Baltimore Railroad. Without counterweights, these engines necessarily had a constant fore-and-aft motion, due to the momentum of the reciprocating parts at each revolution of the wheel. The engine of which Fig. 92 is a section had a middle cylinder of 16⅜ inches in diameter and 18-inch stroke, and two outside cylinders, each of 10¼ inches diameter and 22-inch stroke. The driving wheels were 6 feet 8 inches in diameter, and the whole weight 27 tons.

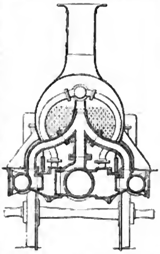

Fig. 92.—Stephenson and Howe's Three-Cylinder Engine.

Another arrangement of engine, of which a few were made by Messrs. Stephenson for the South-Eastern Railway (where they are still working), is shown in Fig. 93. Here circular motion is first given to a cranked shaft, supported in bearings fixed under the boiler, and motion is communicated from this shaft to the driving wheels, behind the fire box, by coupling rods.

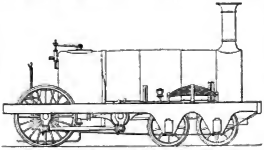

Fig. 93.—Crampton Engine, by Messrs. Stephenson, 1851.

In the early Stockton and Darlington engine (shown in Fig. 45, page 42 of the present work) an intermediate shaft was employed to prevent the pistons (working vertically) from alternately lifting and depressing the engine upon its springs. With the horizontal cylinders, shown in Fig. 93, this necessity does not exist. Mr. William Bridges Adams had, however, employed the arrangement as early as 1848 in his steam carriage "Fairfield," made for the Bristol and Exeter Railway, and it appears also in Mr. Crampton's fifth patent, obtained in 1849, under which patent it is understood the engines shown in Fig. 93 were constructed. This example had 15-inch cylinders, 22-inch stroke, 6-feet driving wheels, and a wheel-base of 16 feet. The weight, in working trim, was 26½ tons, of which 16½ tons bore on the four leading wheels (not grouped in a swivelling truck), and 10 tons on the driving wheels. To say that engines of this kind are very steady on the rails is to say no more than is equally true of many other types of locomotives. On the other hand are obviously increased weight and complication, while, from the position of the driving wheels, they do not support a sufficient proportion of the whole weight of the engine to always insure proper adhesion. With only 10 tons on the driving wheels, their adhesion, estimated at one-sixth of this weight, would be but 3,733 lbs. In addition, therefore, to the cylinder pressure necessary to overcome the internal resistances of the engine, only 54·3 lbs. per square inch could be exerted upon the pistons without slipping the wheels. Thus there were decided disadvantages in the arrangement, with no compensating utility whatever; and for this reason it has not been repeated in subsequent practice.

While the steam pressure was being steadily increased upon English railways between 1847 and 1855, the appearance and introduction of Bourdon's pressure gauge was most timely. The whole subject of steam gauges will be introduced in a further chapter, and it need only be mentioned here that M. Eugène Bourdon, of Paris, devised his gauge in 1849. It was patented in England in December of the same year, in the name of Charles Cowper,* and was for several years made by Mr. Dewrance, formerly of the Liverpool and Manchester Railway. In America it was made and extensively introduced, subsequent to 1851, by Mr. E. H. Ashcroft, of Boston. Its general use has not only given additional security in the working of locomotive and other steam engines, but, as a guide to the engineman and fireman, it has been the means of a considerable saving of fuel, by enabling them to maintain the proper pressure without the former customary indication of a vigorous escape of steam at the safety valves.

In one of his previous patents, Mr. Crampton had specified an engine upon four wheels of equal diameter, and at a considerable distance apart. Midway between the driving axles was a pair of short stout shafts (one on either side of the engine); but, instead of revolving like that shown in Fig. 93, they were made to vibrate like the centre pin of an engine beam, the motion of the piston rods being communicated to these shafts through a stout arm formed on each. At the outer end of each shaft was a pair of opposite arms, of such length that their ends vibrated through a distance corresponding to the diameter of the circle described by the motion of the crank pins in the driving wheels. Coupling rods were then applied from the ends of these arms to the front and hind driving wheels respectively. An engine upon this plan, made as early as 1847 by Messrs. E. B. Wilson and Co., of Leeds, for the Midland Railway, weighed 32 tons; and as there were but four wheels, 6 feet 6 inches in diameter and 16 feet from centre to centre, this was equal to the great weight of 8 tons on a single wheel. Mr. W. H. Barlow has stated that the engine attained a speed of from 75 miles to 78 miles an hour with a light train; but it was only a short time at work. In 1852 one or two engines, upon the same general arrangement, were made by Seth Wilmarth, of Boston, U. S., to the designs of Alba F. Smith, then superintendent of the Cumberland Valley Railroad, upon which line they were employed for a long time with marked success. They had 12½-inch cylinders, 16-inch stroke, and four cast-iron wheels (with chilled rims) 3½ feet in diameter and 16 feet 6 inches from centre to centre.

In 1851 Messrs. Hawthorn obtained an English patent for compensating levers between the wheels of locomotive engines, although the American practice in this respect had been published in the *Practical Mechanic's Journal* for 1850, as well as in the *Franklin Institute Journal* for 1839. Since their first introduction in 1837, by Mr. Joseph Harrison, of Philadelphia, compensating levers or "equalizing beams" have been employed in all American coupled engines, and with great advantage to the working of the machinery, as well as increased ease upon the rails. In the Great Western engines, shown in Fig. 85, page 72, and in the larger Crampton engines, the weight upon the four leading wheels was also distributed, with some advantage, through a single pair of long springs. Means for the same purpose have long been employed in the bogies of American engines and tenders.

Before describing the extensive departures from previous locomotive practice which attended the adoption of unusually steep gradients upon the Vienna and Trieste and the Turin and Genoa lines, space may be properly given to a further record of the progress made by the American engineers. If their designs embodied no new principles, they were at least various in respect of combination and form. In 1846 Septimus Norris, of Philadelphia, proposed the addition of a third pair of coupled wheels to the existing eight-wheel engine. The firm of Norris Brothers, of which he was a member, soon adopted this suggestion for heavy goods engines; but in the meantime Charles Minot, manager of the Boston and Maine Railroad (and now at the head of the Erie Railway), and to whom Mr. Norris had communicated his plans, had two inside-cylinder ten-wheeled engines made by Hinkley and Drury, of Boston, and of these the first was sent from the makers' shops, March 18, 1847. This engine, the "New Hampshire," had 16-inch cylinders, 20-inch stroke, and six coupled wheels 3 feet 10 inches in diameter, besides a truck of four 30-inch wheels. It had 10 square feet of grate area, 56·3 square feet of fire box heating surface, and 141 tubes 1¾ inches in diameter and 10 feet 6 inches long, presenting 678·3 square feet of surface. The same and other makers subsequently made many ten-wheel engines for goods traffic; and this class of engine is still preferred in the States for heavy work; and it is now in use also on the steep gradients of the Great Indian Peninsular Railway (see Plate VI). As compared with the eight-wheeled engine of the American makers, Mr. Norris' plan utilized three-fourths instead of three-fifths of the whole weight for adhesion, while the gross weight was distributed upon ten points instead of eight. As compared with the English six-coupled engine, the ten-wheeled plan, if it had rather less adhesion, was easier upon the rails.

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* Specification, No. 12,889, of patent granted to Charles Cowper, December 15, 1849.

With the exception of the Locks and Canals Company, who copied the "Planet" class of engine as early as 1835, all the American makers kept to outside cylinders until 1845. In May of that year an eight-wheel engine, with inside cylinders, made by Mr. Griggs, of the Boston and Providence Railroad, commenced working on that line. This was the "Norfolk," with 14½-inch cylinders, 18-inch stroke, four coupled wheels of 4 feet 7 inches in diameter, and a truck. The area of the fire grate was 8¾ square feet, the fire box heating surface 44·8 square feet, and the 116 tubes, 1¾ inches in diameter and 9 feet 6 inches long, presented 504·8 square feet of external surface. Weighed by the author in 1853, its working weight was 15 tons 7 cwt. 2 qrs. 20 lbs., of which 9 tons 3 cwt. 2 qrs. 3 lbs. were on the four coupled wheels. The "Norfolk" is believed to have been the first inside-cylinder coupled engine made with a swivelling truck. In December, 1845, Hinkley and Drury, then the leading locomotive makers in New England, sent the "Bangor" passenger engine to the Boston and Maine Railroad, and from that time the American engineers again resorted to inside cylinders, until, by 1855, nearly all had become finally convinced of the advantages of outside cylinders, when placed horizontally, or nearly so, and with counterweights to the reciprocating parts. The "Bangor," with 14-inch cylinders and 18-inch stroke, had four coupled 5-feet 6-inch wheels, the largest in the States at that time, with the sole exception of an outside-cylinder engine, the "Antelope," made by the same makers, and delivered on the Boston and Maine Railroad, September 19, 1845. The "Antelope," with 11½-inch horizontal cylinders and 22-inch stroke, had a single pair of 6-feet driving wheels, besides a four-wheel truck and a pair of small trailing wheels. This engine was evidently intended for high rates of speed, yet it had but 8·1 square feet of grate 43·9 square feet of fire box heating surface, and 80 tubes 10 feet long and 2 inches in diameter, presenting 418·9 square feet of external surface. In 1860 the "Antelope" was working with 5-feet 6-inch driving wheels.

The moderate speed at which American railways are worked obviates the necessity for the liberal heating surfaces given by English engineers. From *Appleton's Railway Guide* for June, 1864, it appears that no express train in America averages more than 32 miles an hour, while on many important lines the average is 26 miles per hour. The interior branch lines are now worked at but two-thirds of this speed, or 17 miles an hour. A movement in favour of increased heating surfaces took place in 1847, the principal addition being made, however, to the tubes. The "Lowell Machine Shop," the former works of the Locks and Canals Company, made two engines, the "Baldwin" and "Whistler," for the Boston and Lowell Railroad, in 1847. These had 15½-inch inside cylinders, 18-inch stroke, four coupled 5-feet 6-inch wheels, and a truck. The fire grate was 8 square feet in extent, the fire box heating surface 56·4 square feet, and 140 tubes, 11 feet long and 2 inches in diameter, gave 806·3 square feet of surface. In 1848 the ten-wheel engines shown in Fig. 50, page 46, were made for the New York and Erie Railroad (of 6-feet gauge), with 18-inch cylinders, 20-inch stroke, and 5-feet driving wheels; they had 14·67 square feet of grate, 72 square feet of fire box heating surface, and 198 tubes 13 feet long and 1¾ inches in diameter, and presenting 1,127 square feet of external surface. At the same time the passenger engines for the same line, with 17-inch cylinders, 20-inch stroke, four coupled 6-feet wheels, and a truck, had 13 square feet of grate, 67 square feet of fire box surface, and 175 tubes 11 feet 6 inches long and 1¾ inches in diameter, presenting 922 square feet of external surface. As in all American engines burning wood, a separate blast pipe was provided for each cylinder, and this at the nozzle was 2¼ inches only in diameter. A prevalent pattern of inside-cylinder passenger engine, then employed on the narrow-gauge roads of New England, had 15-inch cylinders, 18-inch stroke, four coupled 5-feet wheels, and a truck. The area of fire grate was 8·4 square feet, fire box heating surface from 50 to 57 square feet, and with (in various examples) from 123 to 133 tubes 9 feet 11 inches long and 1¾ inches in diameter, presenting from 558·8 to 604·3 square feet of heating surface. The diameter of blast pipe to each cylinder was sometimes as little as $1\frac{11}{16}$ inches, and seldom larger than 1¾ inches. In 1851 an outside-cylinder engine was completed at the Springfield shops of the Western Railroad of Massachusetts, and, with 15¾-inch horizontal cylinders, 26-inch stroke, and a single pair of 6-feet 9-inch driving wheels, there were provided 11¼ square feet of fire grate, 68 square feet of fire box heating surface, and 196 tubes 12 feet 4 inches long and 1¾ inches in diameter, presenting 1,104 square feet. This engine worked with a single 3½-inch blast pipe. Its weight when working was 22¾ tons (or exactly 50,885 lbs). At a public contest at Lowell, already referred to, it drew a weight of 85 tons, besides its tender, over 9 miles of nearly straight and level line in 12 minutes 18 seconds from starting, or at an average rate of 43·9 miles per hour, two consecutive miles having been made at the rate of 50·35 miles an hour. Allowing no more than 25 lbs. per ton for the resistance of engine, tender, and train, rather more than 400 indicated horse-power were exerted at this speed, the corresponding average pressure upon the pistons throughout the stroke being 39·86 lbs. per square inch. The boiler pressure was 84 lbs.

In 1852 passenger engines of 16½-inch cylinders, 22-inch stroke, and four coupled 7-feet wheels were adopted upon the Hudson River Railroad, and two were worked upon the Reading Railroad, with 17-inch cylinders, 2-feet 6-inch stroke, and four 7-feet wheels, and a four-wheeled truck. The last-named engines had large boilers for burning anthracite coal; but, as has been the case with all, or nearly all, large-wheeled American engines, smaller driving wheels have since been substituted for those with which they commenced to work.[*]

The proportions and construction of American locomotives were visibly improved between 1849 and 1855 by the late Thomas Rogers, then at the head of the firm of Rogers, Ketchum, and Grosvenor, of Paterson, New Jersey. Mr. Rogers was not an inventor, but, as a constructing engineer, he was the first to give the authority of a large and successful practice to many of the best features of the American engine of the present day. He increased the former allowance of heating surface, and adopted the link motion,[*] with steam passages of large size, and with 4½ inches to 5 inches maximum travel of the valves. Giving the preference to the outside-cylinder engine, he was among the very first in America to properly counterweight the reciprocating as well as the revolving parts, and he placed the cylinders in a nearly horizontal position, at the same time spreading the truck wheels widely apart. He strengthened the fastenings of the cylinders, and otherwise made the present favourite type superior to the inside-cylinder engines then in use in the States. In its general form, as well as in many points of detail, the Rogers engine has long been closely copied by most American makers. Fig. 94, although from an example by another firm, will sufficiently represent it externally; while Plates XVIII., XIX., and XX. of the present work will illustrate the engine now made by the Rogers Locomotive and Machine Works, Mr. Rogers' former business being continued by his sons, under the engineering direction of Mr. William S. Hudson, once a pupil of George Stephenson's.

It may be added, that, while driving wheels as large as 6 feet only came to be generally employed in American practice subsequently to 1851, from which time Mr. Rogers made many of that size, the general preference in the States has since returned to 5½ feet, and even 5-feet wheels are now thought sufficiently large for the greater portion of the passenger traffic on American railways.

In 1852 a portion of the Baltimore and Ohio Railroad, west of Cumberland, was opened, with 30 miles of gradients of 1 in 45½ (116 feet per mile), and 1 in 50, with frequent curves of 600 feet (about 9 chains) radius. The conditions, therefore, were much the same as were presented upon the Semmering division of the Vienna and Trieste Railway, opened some time afterwards. The older portion, however, of the Baltimore and Ohio Railroad, worked since 1831, had 5 miles of a gradient of 1 in 64½, with curves also of 600 feet radius. Coal-burning goods engines had been worked from the first upon this line; and as early as 1844 these had taken the form shown in Fig. 95, which represents even yet the prevailing class of engines employed not only upon the Baltimore and Ohio, but also upon the Reading Railroad, the latter having a great coal traffic upon gently descending gradients, no-

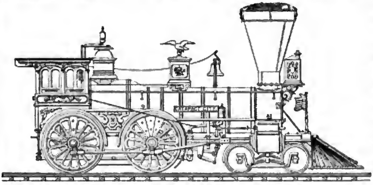

Fig. 94.—American Passenger Engine, by Danforth, Cooke, and Co., 1855.

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[*] The history of large-wheeled engines in America has been a short one. The "Lightning" Crampton engine, made by Edward S. Norris, of Schenectady, in 1849, for the then Utica and Schenectady Railroad, had 16-inch cylinders, 22-inch stroke, and a single pair of 7-feet wheels, but was worked for only a short time. The "Mameluke" inside-cylinder (15-inch by 24-inch stroke) engine, made in 1849 by the Amoskeag Company, had coupled 7-feet wheels; but this and the "Carroll of Carrollton," made by Ross Winans in the same year, and which had a single pair of 7-feet wheels, and a truck under each end, was soon laid aside. Only two engines with 7-feet wheels were made for the Hudson River Railroad, and but two for the Reading Railroad; these have been already noticed. In 1850 Norris Brothers made two outside-cylinder engines with 14-inch cylinders, 22-inch stroke, and coupled 7-feet driving wheels, for the New York and Erie Railroad (now the Erie Railway). Besides the six Crampton engines with 8-feet wheels on the Camden and Amboy Railroad, the nine engines just enumerated were, the author believes, all that were ever made in the States with 7-feet wheels, although two others on the Western Railroad had respectively a single pair of 6-feet 9-inch wheels, and coupled wheels 6 feet 10 inches in diameter. Smaller wheels have been substituted for those of nearly all these engines, as also in the case of the few engines originally made with 6½-feet wheels. This size was only reached in five or six engines on the Hudson River Railroad, three engines on the Harlem, two on the Pennsylvania Central, and one on the Vermont Central Railroad. All the large-wheeled engines had small boilers. With a single pair of driving wheels the adhesion was in all cases insufficient. Winans' engine, "Carroll of Carrollton," had an additional pair of small vertical steam cylinders over the journals of the driving axles, the pistons being made to press down upon the boxes, so that the adhesion weight might be varied between 3 tons and 12 tons.

[*] Mr. H. W. Farley's engine, the "Courier," made by him for the Eastern Railroad in 1847, had the suspended link. This was the earliest application of this form of valve gear in New England.

where exceeding 1 in 264. The object being great tractive force at a slow speed, the Winans engines had large cylinders and small coupled wheels. The earlier examples, or those, at least, made in 1849, had 17-inch cylinders, 22-inch stroke, and 3-feet 10-inch wheels. In 1851 new engines were ordered with 19-inch cylinders, 22-inch stroke, and 3-feet 7-inch wheels, the extreme wheel-base being 11 feet 3 inches. The cast-iron wheels had chilled rims, although separate chilled tyres were afterwards adopted. Only the end wheels were flanged. The fire grate, for coal, was 7 feet long, and 3 feet 6 inches wide, presenting 24½ square feet, the fire box being of the form shown in Fig. 95, and provided not only with doors at the back, but with feeding hoppers at the top. The foot-plate for the two firemen was formed upon the tender, the engine driver riding on the top of the boiler. The latter was 46 inches in diameter, and contained 103 iron tubes 2¼ inches in diameter and 14 feet long. The slide valves could be worked either by an eccentric to admit steam for an average of 16¾ inches of the stroke, or by a cam to cut off at 8⅝ inches and 10 inches of the front and back strokes respectively. The steam ports were 15 inches by 1 inch, and were fully opened by the cam as well as by the eccentric. The weight of these engines was 24 tons 4 cwt. only, averaging 6 tons on each pair of wheels. In 1852-53 the Baltimore and Ohio Railroad Company's Master of Machinery, Samuel J. Hayes, constructed five engines much like that shown in Fig. 95, but with 20-inch cylinders, 22-inch stroke, and eight coupled wheels 3 feet 7 inches in diameter. The weight of these, in working order, is 25 tons 12 cwt. 2 qrs. The regular load of these engines, on a continuous gradient of 1 in 45½ for 11 miles, is eight wagons, each on eight wheels, and weighing, with load, 114 tons (128 American tons of 2,000 lbs). The 20-inch cylinder engines have often taken up eighteen of the same wagons, or a total load of 200 to 205 tons, at a speed of 8 miles an hour. Taking one instance where the gross load was 201 tons, the performance may be analysed as follows:—

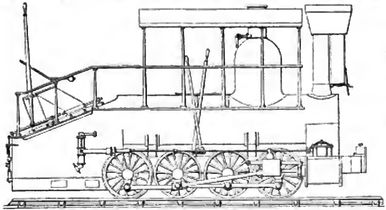

Fig. 95.—Ross Winans Coal Burning Goods Engine, 1844 to 1854.

Total weight, engine, tender, and train, . . .	242 tons.
Gravity on incline of 1 in 45½ (116 feet per mile),	11,912 lbs.
Friction, at an average of say 10 lbs. per ton, .	2,420 ,,
Total resistance of engine, tender, and train, and consequent tractive force of engine, . .	14,332 ,,

Equal to one-fourth the weight on the driving wheels, and equal to a mean pressure of 70 lbs. per square inch on the 20-inch pistons throughout their strokes.

A series of trials was made in the autumn of 1852, and in January, 1853, with two of the 19-inch and two of the 20-inch cylinder engines.

Date of trip, ...	Oct. 21, 1852.	Oct. 22, 1852.
Trip from Piedmont to Altamont, .	11 miles up 1 in 45½.	
Number of engine, ...	No. 83.	No. 103.
Diameter of cylinder, ...	20 in.	19 in.
Stroke of piston, ...	22 in.	22 in.
Diameter of eight coupled wheels, .	43 in.	43 in.
Weight of engine, ...	37,400 lbs.	34,200 lbs.
Number of wagons drawn, . . .	8	8
Weight of engine, tender, and train, .	126¾ tons.	127¾ tons.
Resistance of gravity of this weight on gradient, . . .	6,236 lbs.	6,284 lbs.
Whole resistance, friction estimated at 12 lbs. per ton, . . .	7,756 lbs.	7,816 lbs.
Average speed in miles an hour, . .	11·46	14·37
Equivalent velocity in feet per minute,	1,008½	1,264½
Horse-power exerted, . . .	237	299½
Equivalent average pressure on pistons throughout their stroke, .	37·9 lbs.	42·3 lbs.
Boiler pressure, ...	80 lbs. to 106 lbs.	90 lbs. to 122 lbs.

The following trips were made up a gradient of one in 64·4, 2½ miles long, at Parr's Ridge, east of Harper's Ferry, with one of the same engines, "No. 83," and with a like engine, "No. 72."

	Engine No. 72.						Engine No. 83.			
Date of trial, Nov., 1852, and Jan., 1853, . .	Nov. 10.	Nov. 13.	Nov. 17.	Nov. 20.	Nov. 24.	Nov. 29.	Jan. 7.	Jan. 11.	Jan. 18.	Jan. 21.
Number of eight-wheel loaded wagons drawn, .	18	16	17	17	17	17	16	18	17	14
Weight of engine, tender, and train, . . tons,	257	238	234½	254	245	248	230½	283	239½	222¼
Gravity and friction of train, the friction at 10 lbs. per ton, ... lbs.,	11,511	10,657	10,502	11,385	10,978	11,104	10,332	12,673	10,725	9,964
Average speed in miles per hour, . . .	7½	12½	10	7·89	8·82	8·82	10·71	10·34	9·37	13·04
Equivalent velocity in feet per minute, . .	660	1,100	880	694	776	776	942·4	910	824½	1,147½
Horse-power exerted, ...	230·2	355	280	239·4	258	261	294¾	349½	268	346·4
Equivalent average pressure on 20-in. pistons throughout their stroke, . . . lbs.,	56¼	52	51·3	55⅞	53¼	54¼	50·4	61·9	52·4	48·7
Boiler pressure, ...	...	100 lbs. to 125 lbs.			...	...	100 lbs. to 125 lbs.			...

A series of experiments was also made upon two of the Winans engines, Nos. 102 and 118, up the same gradient of 1 in 64·4. These, with their results, were as follow:—

	Engine No. 102.					Engine No. 118.			
Date of trial, Nov., 1852, and Jan., 1853,	Nov. 11.	Nov. 15.	Nov. 18.	Nov. 22.	Nov. 26.	Jan. 4.	Jan. 8.	Jan. 26.	Jan. 29.
Number of eight-wheel loaded wagons drawn,	18	16	15	17	17	18	18	17	17
Weight of engine, tender, and train, . . . tons,	252¼	236·7	236	245·2	239¾	262¼	287·3	245·2	228¾
Gravity and friction. Friction, 10 lbs. per ton, . . lbs.,	11,296	10,599	10,569	10,980	10,736	11,766	12,865	10,980	10,243¼
Average speed, miles per hour,	7·89	4·05	3·49	Not taken.	Not taken.	9·68	7·50	8·82	13·63
,, ,, feet per minute,	694	356·4	307	,,	,,	852	660	776	1,199·4
Horse-power exerted,	237½	114·4	98·3	,,	,,	303¾	257·3	258·2	372·3
Corresponding mean pressure on 19-inch pistons, . lbs.,	61·1	57¾	57	59·4	58	63·7	69¾	59·4	55·4
Boiler pressure,	100 lbs. to 145 lbs.								

The trials on the 2½-mile gradient of 1 in 64·4 (and which were rather beyond the average working of the line) occupied each but 10 or 15 minutes, so that the corresponding evaporation could not be correctly estimated. But the gross evaporation in these engines was observed over a journey of 98 miles, and although the time lost in waiting at stations was nearly equal to that occupied on the several trips, a conclusion may be fairly drawn. The average of six trips of one of the 20-inch cylinder engines gave an evaporation of 712½ cubic feet for 9 hours 2 minutes' running and 6 hours 24 minutes' standing time. Dividing the evaporation over the running time, it would be 79 cubic feet only per hour, the evaporation of water in this case being 9·04 lbs. per pound of coal. As, however, in the other experiments, from 5·16 lbs. to 8·5 lbs. of water were evaporated per pound of coal, it is possible that, in the first-named experiment, some of the water was worked through the cylinders without having been evaporated at all. About 125 cubic feet of water per hour of running time was the maximum evaporated in any of the trials, while in several the rate of evaporation was below 70 cubic feet per hour, exclusive of waste in stoppages. During the short run up the steep gradient, the consumption of water as steam was at the rate, probably, of 125 to 200 cubic feet per hour. The 20-inch cylinder engines, by Mr. Hayes, had 18 square feet of grate, 87½ square feet of fire box heating surface, and 134 iron tubes 2 inches in diameter and 14 feet long, presenting 984 square feet.

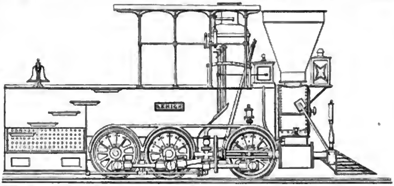
Fig. 96.—Colburn's Engine for Burning Anthracite Coal, 1854.

In 1853 Mr. Winans made two engines (for the 6-feet gauge) which, with hardly a larger boiler than that of the 19-inch cylinder engines shown in Fig. 95, had cylinders of the great diameter of 22 inches, the stroke being also 22 inches, and the diameter of the eight coupled wheels 3 feet 7 inches. The object was, that of working at a high degree of expansion. In 1854 the same maker made an experimental passenger engine, the "Centipede," which was tried upon the Baltimore and Ohio Railroad, and which had 22-inch cylinders, 22-inch stroke, eight coupled wheels 3 feet 7 inches in diameter, besides a swivelling truck in front, making twelve wheels in all, or twenty, when those under the tender were included. Such excessive diameters of cylinder, without corresponding boiler power, were found objectionable; and in some cases, indeed, even 16½-inch cylinders were diminished, by shrinking in a cast-iron lining, to 14¾ inches, and 15-inch cylinders to 13¾ inches. The 17-inch cylinders (30-inch stroke) of two passenger engines on the Reading Railroad were diminished in this way to 15 inches; but in nearly every instance, the author believes, the bushings have been since removed, and the cylinders worked at their original diameter.

Down to 1855, and even later, brazed copper tubes were generally employed in American wood-burning engines. As wood, even under a strong blast, does not cut away the tubes in the manner of hard, flying particles of coke, copper was found to answer very well. The coal-burning engines made from 1831 to 1857 almost invariably had iron tubes. The manufacture of Green's solid drawn brass tubes was introduced into the States in 1851 by Mr. Fox, now of Messrs. Fox Brothers, of Derby (England). The American Brass Tube Works, at Somerville, near Boston, were planned and started by Mr. Fox, acting for a company formed under the auspices of the proprietors of the Boston Locomotive Works. Brass tubes have since attained to extensive use in the States.

In 1854 the author, then engineer to the New Jersey Locomotive Works, at Paterson, designed and constructed a number of anthracite coal-burning engines for the 6-feet gauge. Mr. Winans' practice in burning anthracite was already successful, and in many respects it was followed in the class of engines shown in Fig. 96. These, although their boilers were designed for 20-inch cylinders, were made with 18-inch cylinders, 24-inch stroke, and six coupled wheels 4 feet in diameter. The wheel-base was 11 feet only, to pass through short curves in branch lines leading to the coal pits. The boiler, 50 inches in diameter, had 91 iron tubes 3 inches in diameter and 15 feet 6 inches long—a length which permitted the fire box to be placed entirely behind the driving wheels, and thus a clear width of 7 feet 6 inches was obtained for the fire grate, the bars being 6 feet long. This enormous grate area, 45 square feet, was found useful in burning so refractory a fuel as anthracite. With a large blast pipe, the tube surface, 1,008 square feet in extent, was found ample to take up the heat generated. The spreading of the fire box to a width greater than that of the gauge of the line,* by placing the fire box entirely behind the wheels, had been carried out by the author in a number of 6-ton tank engines which he designed and made, early in 1852, for a contractor's line of 3-feet 3-inch gauge, that of the permanent way (the Great Western Railway of Canada) being 5 feet 6 inches. In the engine shown in Fig. 96 the overhanging weight at the back was no doubt considerable, the cast-iron grate bars alone weighing upwards of 1¼ tons. There was, however, great weight forward, the rough weight of each cylinder casting being 19¾ cwts., while a cast-iron "crypt," to the walls of which the cylinders were fastened with great strength, weighed 14 cwt. The middle axle was nearly under the centre of gravity of the engine, and this axle having stiffer springs than the others, the whole weight of 33 tons was very equally distributed. Fig 97 shows a transverse section through one cylinder, and one of the corbels supporting the guide bars.

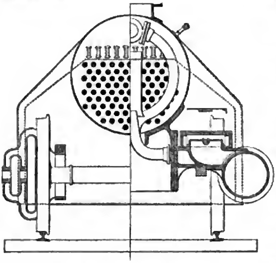

Fig. 97.—Colburn's Engine, Section.

During the construction, in 1852, of a part of the Baltimore and Ohio Railroad, a temporary line (over one of the tunnels then in progress) was successfully worked by one of the 19-inch cylinder eight-coupled Winans engines, which regularly ascended an incline slightly steeper than 1 in 10, or 530 feet per mile, drawing behind it its tender and an eight-wheel loaded wagon weighing 13 tons. The engine weighed 24¼ tons, all on the coupled wheels, but upon an incline of 1 in 10 only nine-tenths of its weight, or say 21¾ tons, could press effectively upon the rails. The tender weighed 12¾ tons, and the whole moving mass of 50 tons must have opposed a resistance due to gravity of 11,200 lbs. Adding 650 lbs., at 13 lbs. per ton, for friction, we have a total resistance of 11,850 lbs.; and as the ascent was often made at the rate of 10 miles an hour, or 880 feet per minute, the corresponding work done was 316 horse-power, and the necessary effective mean pressure upon the pistons 64 lbs. on the square inch. Occasionally, during frosty weather, the engine, tender, and wagon slipped from the top to the bottom of the gradient, showing the adhesion to have been then less than one-tenth of the weight.

Early in 1854 the late Mr. Charles Ellet, then engineer to the Virginia Central Railroad, commenced working a short line of heavy gradients and sharp curves over one of the tunnels then in progress through the Blue Ridge. This line was 4 miles 3 furlongs long, and rose on the eastern side 610 feet in 2 miles 3 furlongs, and fell on the western slope 450 feet in 2 miles, giving an average rise of 1 in 20½ on the eastern, and 1 in 23½ on the western slope. The maximum gradient, however, was 1 in 17·84, or 296 feet per mile, while there were several curves of 300 feet (4½ chains) radius, and in one case the radius was but 234 feet (say 3½ chains). The gradient was flattened, however, to 237⁶⁄₁₀ feet per mile, or 1 in 22·22, upon both the 300-feet and the 234-feet curve, and when the flanges of the engine wheels were lubricated, the resistance upon the straight incline of 1 in 20½ was found to be the same as on the 300-feet curve, inclined at 1 in 22·22. This line was regularly worked for some years, with passenger and goods traffic, by tank locomotives, made by M. W. Baldwin and Co., and of the following dimensions and arrangement:—Outside cylinders, 16½ inches in diameter, 20-inch stroke, and six coupled wheels 3½ feet in diameter, with a wheel-base of 9 feet 4 inches. The whole weight of the engine, with 100 cubic feet of water, and 100 cubic feet of wood, was 55,000 lbs., or rather more than 24½ tons. The loads taken over the mountain were, for passenger traffic, two 8-wheel passenger carriages and one 8-wheel luggage van, while goods trains of three 8-wheel wagons were regularly worked. The extreme load was 45 tons, taken up at the rate of 7½ miles per hour, the descent being made at a slower rate, say 5½ to 6 miles an hour. Trains of 38 tons weight were more usual. On a gradient of 1 in 18, the total resistance thus opposed by gravity would be 7,785 lbs., and if 12 lbs. per ton be allowed for the friction of the engine and train at this slow rate of speed, the total resistance to be overcome is 8,540 lbs., corresponding to a mean effective pressure of 66 lbs. per square inch of the pistons. The motion of the engine and train being at the rate of 660 feet per minute (the mean speed of piston being 200 feet per minute) the work done would be 170½ horse-power.

In an account of the *Mountain Top Track* published by Mr. Ellet in 1856, he said, "To enable the engines better to adapt themselves to the flexures of the road, the front and middle pair of drivers are held in position by wrought-iron beams, having cylindrical boxes in each end for the journal bearings, which beams vibrate on spherical pins fixed in the frame of the engine on each side, and resting on the centres of the beams. The object of this arrangement is to form a truck, somewhat flexible, which enables the drivers more readily to traverse the curves of the road." (The "drivers" are the coupled driving wheels.) The cylindrical boxes here referred to were described under Mr. Baldwin's patent of August 17, 1835.[*] Means for coupling the driving and truck wheels were described in another of his specifications filed in pursuance of a patent of December 31, 1840.[†] (In the same specification was described the making of the "pedestals," or axle guards of locomotive engines, solid with the frames, as in the present English and American practice.) In his specification of a patent dated August 25, 1842,[‡] Mr. Baldwin described the combination of two pairs of axles, having cylindrical boxes, with the "beams" referred to in Mr. Ellet's description; and in his subsequent practice the patentee coupled the wheels of such trucks directly with the driving wheels, by means of coupling rods. This arrangement is shown in Figs. 98 and 99. As the truck, in passing through curves, worked upon the beams described by Mr. Ellet, the truck axles, although they remained parallel to the driving axle, took new positions upon the rails, as best shown in Fig. 99. The coupling rods were made with cylindrical brasses, to enable them to accommodate themselves to the varying positions of the "truck" axles. Geometrically, no doubt, this combination of wheels could only work properly around curves by a lengthening and shortening of the rods which served to couple the principal pair of driving wheels with the hind "truck" wheels. But if the coupling rods from the principal pair of driving wheels be 5 feet long, and if the beams of the truck frame be 4 feet long (the radius of the curve described by the axle boxes around the spherical side bearings being 2 feet), then the total corresponding lengthening of the coupling rods in order to allow the hind truck wheels to move 1 inch to one side, and the front wheels of the truck 1 inch to the other side of their normal position on a straight line would be

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* The author is informed that this had been done still earlier in an engine made at Wolverton, and known as the "Mangle."

$$\sqrt{60^2 + 1^2} - 60 + 24 - \sqrt{24^2 - 1^2} = 0{\cdot}0275 \text{ inch},$$

or less than $\frac{1}{32}$d of an inch. And, if only one pair of driving wheels were thus coupled with a four-wheel truck, the total wheel-base being 9 feet, the motion permitted by this slight elongation of the coupling rods (an elongation provided for by a trifling slackness in the brasses) would enable the three pairs of wheels to stand, without "binding," in a curve of only 100 feet radius. This mode of enabling coupled engines to pass through short curves has been practised by Mr. Baldwin for many years, and it has been lately adopted, with changes of detail, by M. Beugniot for certain of the French lines.

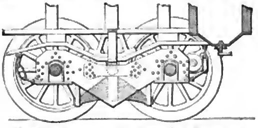

Fig. 98.—Baldwin's "Truck" for Coupled Goods Engines, 1842.

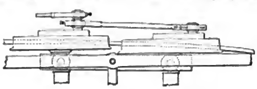

Fig. 99.—Half Plan of Baldwin's "Truck."

So lengthy a record of American practice is, perhaps, justified by the fact that it was greatly influenced at an early period by the necessity for working the steep gradients of American railways, and that the experience thus obtained was avowedly consulted by the engineers who laid out the Semmering section of the South Austrian line.[*] On the Hudson and Berkshire Railroad a gradient of three-fourths of a mile, rising at the rate of 154½ feet per mile, or 1 in 34·17, has been worked by locomotives for twenty-five years. On the Western Railroad of Massachusetts a continuous gradient of 7½ miles rises at the rate of 78 feet per mile, or 1 in 67·7. This was laid out by Major Whistler (afterwards engineer-in-chief of the St. Petersburg and Moscow Railway), and it has been regularly worked by locomotives since September, 1841. A short distance from it is a gradient rising for a mile at the rate of 1 in 64. Gradients rising at the rate of 1 in 88 for several miles in length were adopted and worked in 1840 upon the New York and Erie Railroad, and from that time the performances of the Norris engines upon the Lickey incline, of 1 in 37, on the Birmingham and Gloucester Railway, were constantly quoted in the States as a sufficient precedent for the adoption, where necessary, of gradients of even greater steepness.

The Austrian Government offered a series of prizes, in 1852, for engines each of which should draw a gross load of 110 tons, in addition to its own weight and that of its tender, at the rate of 10 miles an hour, up the 13½ miles on the north side of the Semmering, the average rise being 1 in 47, and the maximum for 2½ miles 1 in 40. The line has curves of a minimum radius of 9½ chains. No English maker competed, and an engine, the "Bavaria," made by Maffei, of Hirschau, near Munich, won the principal prize. In this engine the wheels of the tender were coupled with the others by means of an endless chain, but although great adhesion was thus obtained, this mode of coupling had to be soon abandoned. Mr. Wilhelm Engerth, locomotive engineer to the Austrian Government, then constructed the tender in such manner that its side frames extended forward and past the fire box, a portion of the weight of which they were made to support, and he was thus enabled to place one axle of the tender in *front* of the fire box. The point at which the engine exerted its draught upon the tender was only a short distance behind the driving axle, and, according to the curves of the line, the tender could move radially around this point. Previous to the actual opening of the Semmering line, Engerth's engines had spur gearing to drive the front tender axle from the driving axle, and the tender wheels were coupled by rods working upon pins in outside cranks. In this way the adhesion of ten wheels instead of six, or of 55¼ tons instead of 38¾, was obtained. Since 1853, however, the gearing to the front tender axle has been but little employed, the engines working by the adhesion only of their six coupled wheels. Twenty-six goods engines upon Engerth's system were of the following dimensions and weight:—Outside horizontal cylinders, 18·7 inches in diameter, 24-inch (0·610 mètre) stroke, six coupled wheels 3 feet 6 inches in diameter, with a wheel-base of 11 feet 3¾ inches. The area of fire grate was 12·6 square feet, and that of the fire box heating surface 75 square feet. The boiler contained 189 tubes 15 feet 7 inches long and 2 inches in diameter, and presenting 1,585 square feet. Boiler pressure, 114 lbs., corresponding to the French rating of 8¾ atmospheres. The weight of the engines is not quite the same for all. M. Desgrange, the engineer to the South Austrian Company, gives the weight of one in working order as follows:—Leading axle, 13½ tons; second axle, 11¼ tons; driving axle, 11½ tons; making 36¼ tons for the engine. The weight of the tender, as may be inferred from the description already given of its construction and working, is most unequally distributed, being 3½ tons only for the front axle, while the hind wheels press upon the rails with the enormous weight of 17¾ tons! M. Flachat some time since had the original spur gearing and coupling rods for the tender restored to one of these engines, which was then taken to Vienna and weighed, with the following result:—Leading axle, 15 tons; second axle, 10½ tons; driving axle, 15 tons 8 cwt.; making nearly 41 tons for the engine alone. The weight on the two tender axles was 6 tons 3 cwt. on the front, and 19 tons 14 cwt. on the hind axle, the latter being probably the greatest load ever placed upon a single pair of railway wheels. The whole weight of engine and tender (and all of which was employed for adhesion) was 66½ tons. The tender, it should be added, was made to carry from 5 tons to 7 tons (!) of coal, and upwards of 1,200 gallons of water. The Engerth engines, as introduced twelve years ago upon the South Austrian Railway, were conspicuously defective, and they are being rapidly superseded. In September, 1860, one of these engines, altered in accordance with the directions of M. Flachat, commenced working. It had a fourth pair of wheels coupled with the driving wheels, the tender wheels being behind the fire box. The whole weight of engine and tender was 65 tons, of which the leading wheels bore 11 tons 17 cwt., the second pair 11¼ tons, the third 11 tons 4 cwt., and the trailing wheels 11½ tons, making 45 tons 16 cwt. for the engine. The tender, the wheels of which were not coupled for adhesion with those of the engine, was on two axles, the weight on the first being 8 tons 18 cwt., and on the back axle 10½ tons, the whole weight being 19 tons 8 cwt. Within two years five other engines had been remodelled to conform to M. Flachat's designs, and it is not unlikely that the peculiarities of the Engerth system will soon be abandoned by the Austrian engineers. The old engines, working without the coupling of the wheels of the tender, took up 125 tons of additional weight, at a mean speed of 10 miles an hour, thus performing about the same work as the Winans engines and tenders, of rather more than one-half the weight, upon continuous gradients of 1 in 45½ upon the Baltimore and Ohio Railroad. The engines altered by M. Flachat now take up 172½ tons of load at 10 miles per hour. The whole weight of the train (175 French tons), being 392,000 lbs., and that of the engine and tender 145,300 lbs.: the gravity of this mass upon an incline of 1 in 40 is 13,432½ lbs., and, if we add 12 lbs. per ton for friction, we have 13,432½ + 2,850 = 16,282½ lbs. total tractive effort, at 880 feet per minute, or rather less than 400 indicated horse-power. The average gradient, however, is but 1 in 47, and consequently the mean indicated power for a period of an hour is correspondingly less. The cost of repairs of the Engerth engines was, for 1860, 1·576 francs per kilomètre, or 2s. per mile. This, under M. Flachat's system, will, there is reason to believe, be diminished to 1s. per train mile.

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[*] Vide *Journal of the Franklin Institute*, second series, vol. xvii. (1836), p. 273.

[†] Ditto, third series, vol. iii. (1842), p. 143.

[‡] Ditto, third series, vol. xvi. (1848), p. 93.

[*] "Before commencing the construction of the Semmering Pass of the Southern line, the Austrian Government sent over the engineer of the railway to America, to investigate the subject of steep gradients. On his return the surveys were commenced, and the present line, along the valley of Reichenau, and through Klamm, &c., was selected." Vide *Minutes of Proceedings of the Institution of Civil Engineers*, vol. xv., Mr. Drysdale's paper "On Steep Gradients of Railways." The continuous gradients of 1 in 45½ upon the Baltimore and Ohio Railroad were determined upon, and the works commenced in 1849, and opened in 1852. The railway over the Semmering was opened in May, 1854.

It should be recorded here that the railway saloon carriages of the Austrian imperial court weigh 22 tons, and are borne upon eight wheels, the extreme axles of which are parallel, and at a distance from each other of 26 feet 10¼ inches; and that these are taken over the Semmering at the rate of 18¾ miles per hour, through curves of 9¼ chains radius. The engines improved by M. Flachat take 172½ tons of load up the incline at the rate of 10 miles per hour, with a consumption of only 62 lbs. of coke (or its equivalent of lignite) per mile.

The Engerth system was adopted, with modifications from the original, in the construction, in 1856, of a number of very large goods engines for the Northern and the Eastern Railway of France. For the first-named line forty engines of the kind shown in Fig 100 were made by various makers, among whom were Messrs Schneider, of Creuzot. In these the whole weight of the engine is supported upon its own coupled wheels; but the side frames of the tender are carried forward and past the fire box, and the engine and tender are coupled together just in front of the trailing axles. Otherwise these engines are only remarkable from their great size. They weigh 40 tons 13 cwt. in working trim, the tender (really forming a component part of the engine) weighing, when loaded, 21 tons 7 cwt. more, the total weight being 62 tons. The cylinders (shown in section in Fig. 100) are one-half mètre, or 19$\frac{11}{16}$ inches in diameter, and the stroke 26 inches. The coupled wheels are 4 feet 1⅝ inches in diameter. The inside diameter of the boiler is 4 feet 11 inches, and it contains 235 tubes 2⅛ inches in external diameter and 16 feet 4¾ inches long, clear of tube plates. The grate is 4 feet 8¾ inches long and 4 feet 5 inches wide, presenting 20·93 square feet, the fire box heating surface being 104½ square feet, and the external tube surface 2,219½ square feet. The height of the centre of the boiler above the level of the rails is 6 feet 5½ inches. With 4 inches of water over the crown plate, there are 171½ cubic feet of water room and 72½ feet of steam room. The boiler pressure is 103 lbs., or 8 atmospheres. The blast pipe is variable from 6 to 27 square inches of area. The steam ports are 13¾ inches by 1¾ inches, the exhaust port to each cylinder being 13¾ inches by $3\frac{1}{16}$ inches. The valves have $1\frac{1}{16}$-inch outside and ⅛-inch inside lap, with ⅛-inch lead, and the admission is variable from 10 per cent. to 79 per cent. of the stroke. The engine wheel-base is 13 feet, and the tender wheel-base 5 feet 7 inches, while the intermediate space from front wheels of tender to hind wheels of engine is 10 feet, making 28 feet 7 inches in all. The weight is very equally distributed, the least on any one pair of wheels (the first pair from the front wheels) being 9 tons 13 cwt., while the greatest weight (on the hind tender wheels) is 11 tons 2½ cwt. These engines, working upon a gradient of 1 in 200, 15 miles long, draw, according to M. Flachat, 561½ tons (English), exclusive of engine and tender, at a mean speed of 10½ miles per hour. M. Flachat estimates the resistances at the rate of 9½ lbs. per ton, besides that of gravity, equal to 11·2 lbs. per ton. The total resistance overcome, at 20·7 lbs. per ton for 623½ tons, is 12,907 lbs., and at 10½ miles an hour, or 924 feet per minute, this represents 361½ horse-power, the mean effective pressure upon the pistons being 64 lbs. nearly. These engines, M. Flachat reports, evaporate 9·2 lbs. of water per pound of coal, a rate due, very obviously, to the great proportionate extent of the tube surface.

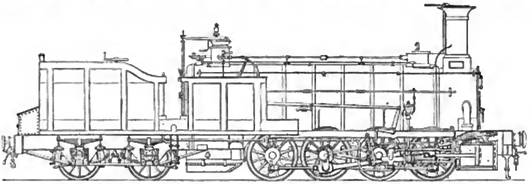
Fig. 100.—Engerth Goods' Engine, Nord Railway of France, 1856.

The coupling of the tender to the engine at a point in front of the fire box, by extending the framing of the former some distance in front of the latter, is the special feature of the Engerth system; and it is relied upon in order to allow the connected engine and tender to pass freely through short curves. The same end was accomplished twenty years ago, in the Winans engines, Fig. 95, by means of a long draught bar, or "pulling bar," upwards of 10 feet long and 3½ inches in diameter, passing through the ash pan, and coupling the engine and tender together in front of the fire box, although equal freedom of movement was permitted around the coupling pin at that end of the draught bar under the front end of the tender.

It will be as well, while upon the subject of heavy goods engines on the Continental Railways, to at once follow up the recent changes which have given to the French locomotive of the present day so remarkable a character for size and arrangement. This will require that the notice of other features of the locomotive practice of 1850-60 be temporarily deferred; but a more complete survey may thus be taken of broad and distinct fields of procedure. It will be seen that the French practice is as various as it is remarkable.

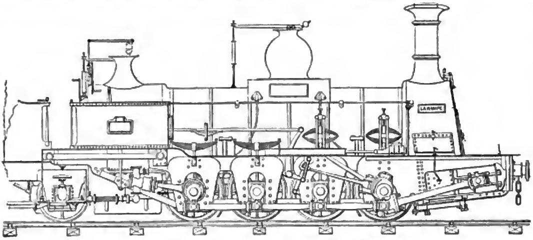
Fig. 101.—Bougniot's Goods Engine, by Koechlin, 1860.

M. Beugniot's engines* were designed for greater tractive force and flexibility of motion than were possessed by the heavy Engerth engines just described. Messrs. Kœchlin, of Mulhouse, made two engines upon M. Beugniot's plans, in 1860, and these, "La Rampe" and "La Courbe," were for some time worked upon the Paris, Lyons, and Mediterranean Railway. One of these is shown in Fig. 101, which shows also one of the tender wheels, a small portion of the weight of the engine being made to rest upon the tender, to partly neutralize the vertical action of either in passing over slight obstructions on the line. The cylinders, 21¼ inches in diameter (the stroke of piston being 22 inches), were placed 5 feet 1¼ inches from centre to centre, and directly in front of the leading wheels, which were the driving wheels of the engine. The piston rods came through the front cylinder covers, and by means of stout cross-heads each worked the driving wheels through a pair of connecting rods turning respectively an outside crank and a crank in the driving axle, as shown in a half-plan through one cylinder in Fig. 102. It is not easy to discover what precise advantages were sought in this roundabout arrangement, unless it were an object to display the indisputable ingenuity of the designer. The axles of the four pairs of coupled wheels were provided with both outside and inside axle boxes, each having a bearing spring. The outer axle boxes had a lateral play of 1 inch in the axle guards; and that part of the weight borne upon the inner boxes was applied through swivelling beams, acting in the manner of Baldwin's truck, Fig. 98. The leading axle, which was also the driving axle, had in addition a fifth axle box at the middle of its length. The wheel-base—with driving wheels 3 feet 11¼ inches in diameter—was 12 feet 9½ inches only; and thus, when the peculiar arrangement of the running gear is considered, a good degree of flexibility was obtained for working through curves of small radii; yet the plan was not one likely to be generally adopted. With a total heating surface of 1,861½ square feet, the weight of the engine was 46 tons 11¼ cwt., while the total weight of engine and tender was 69¾ tons. The extreme width of the engine was 9 feet 4 inches, or beyond the limit usually permissible upon the narrow gauge. The cylinders were supported only by the external framing, and by the front and back plates of the smoke box. M. Nozo, of the Chemin de Fer du Nord, has stated that these engines have exerted a force, including that necessary to overcome their own friction, of 16,030 lbs., at the rate of 8·95 miles an hour, or 787½ feet per minute; equivalent to 382½ horse-power. A tractive force of 16,537½ lbs. has been sometimes exerted.

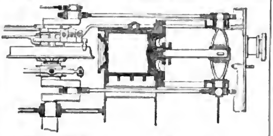

Fig. 102.—Half Plan Through Cylinder, Beugniot's Engine.

In 1860 M. Flachat published an elaborate memoir upon a proposed railway over the Alps. To avoid tunnels, the line was to be taken over St. Gothard with gradients of 1 in 20, and with curves of 66 feet radius. For the work M. Flachat proposed a boiler carriage of great size, with small cylinders and driving gear sufficient only for its own propulsion, while the tender and carriages were each to be provided with engines of moderate power, to be supplied with steam through a suitable pipe leading from the boiler. Fig. 103 is a longitudinal, and fig. 104 a transverse section of the boiler carriage. The boiler was to have 3,981 square feet of heating surface, and of itself was to weigh 18¼ tons, while the water carried in it would weigh 6½ tons more. The whole weight, including framing, working parts, &c., was to be 40 tons, and with tender 62 tons; and it was estimated that 142 tons of additional weight might, with the supply of steam thus commanded, be taken up long gradients of 1 in 20. Each axle was to have but one wheel, all the wheels upon one side of the engine and carriages thus acting independently of those upon the other. The plan of a pair of axles for each pair of wheels was that patented in 1826 by Robert Stephenson. As applied to locomotive engines, it was attended with the risk that, instead of the cranks on opposite sides remaining always at right angles to each other, they might occasionally—in consequence of slight differences of diameter of the driving wheels, or by slipping—acquire the same angular direction with reference to their axles. With a pair of cylinders, however, under each end of each eight-wheel carriage, the cranks of all the wheels under the train could hardly acquire the same angular position, so as to cause a difficulty in starting or continuing

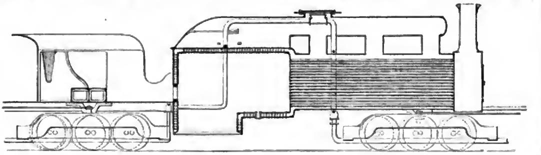

Fig. 103.—Flachat's Alpine Railway, Stock, 1860.

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* Patented in England in the name of J. Henry Johnson, November 29, 1858.

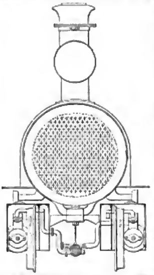


Fig. 104.—Flachat's Boiler Carriage, 1860.

the motion of the train. It would appear, however, that much of the steam carried through long pipes, to a distance from the boiler, would be condensed in an Alpine climate. M. Flachat's design was part of a comprehensive scheme for the construction and working of railways in mountainous countries, and it may yet be brought to the test of conclusive trial. Already, on the Great Northern Railway, cylinders and driving gear have been fitted to the tenders of a considerable number of goods engines, and steam is supplied to work them by a copper pipe leading from the boilers. The spent steam from the cylinders under the tender is not available for the purposes of draught in the chimney of the engine, but it is employed to heat the feed-water, although a temperature of much above 100° effectually prevents the use of the Injector.

Next in order among remarkable goods engines for steep gradients and sharp curves is the "Steierdorf," exhibited in London in 1862, and represented in Plate XXI. The peculiarity of this is in the mode of coupling the engine and tender wheels. This, although well shown in the plate, will be more fully described in a further chapter.

The latest example of the modern spirit of bold departure from established models is shown in the goods engines designed by M. Petiet, engineer in-chief of the Northern Railway of France, and constructed for that line by MM. Gouin et Cie., of Paris. Fig. 105 is an elevation, and Figs. 106, 107, and 108 are transverse sections, of these engines, of which a considerable number arè now working. They are on twelve wheels, grouped in sets of six each, each set being worked by a separate pair of cylinders. The diameter of the four cylinders is 17¼ inches, the stroke of the pistons is the same, and the diameter of the coupled wheels is 3 feet 6 inches. The extreme wheelbase is 19 feet 8½ inches, the six axles being parallel with each other. The whole weight of the engine, with 7 tons 17½ cwt. of water and 2 tons 7 cwt. of coal in the tank, is 59 tons, or rather less than 10 tons to each pair of wheels, the distribution of weight upon which is very equal. The boilers of these engines are of unusual size, and the steam-drying apparatus and chimney are novel in their arrangement. The fire grate is 6 feet ¾ inch long, and 5 feet 10⅝ inches wide, being placed quite above the wheels (which run upon the 4-feet 8½-inch gauge). The area of the fire grate is 35·84 square feet—greatly exceeding that for any locomotives previously constructed, with the exception of those shown in Fig. 96. The height from the fire grate to the crown plate is 3 feet 9⅝ inches at the back, and 4 feet 4 inches at the tube plate; and the whole area of the fire box heating surface is 108 square feet. The boiler is 4 feet 10 inches in external diameter, and contains 464 tubes 1¹¹⁄₁₆ inches in external diameter, and 11 feet 5⅝ inches long, the external tube surface being 2,201 square feet. The waste heat, however, instead of passing directly into the chimney, traverses a tubular steam chamber, thereby drying the steam. The area of this "dryer" is 154 square feet. The chimney and blast pipe are horizontal, this position answering every purpose with an artificial draught. The chimney is turned up at its end, to prevent discharging the products of combustion directly over the train. The boiler pressure is 118 lbs. per square inch. One of these engines is reported (by M. Brüll) to have drawn a train of twenty-one loaded wagons, weighing 263 tons, up a gradient rising 1 in 55¼ for 2¼ miles, at a mean rate of nearly 10 miles an hour. The total resistance of friction and gravity for the 322 tons of engine and train is estimated by M. Brüll as 18,747 lbs., the horse-power, varying according to the speed, being from 472 to 629. In the description of these engines furnished by the Northern Railway Company to the jurors in Class V. of the International Exhibition of 1862, the diameter of the cylinders was given as 16³⁄₁₆ inches. The

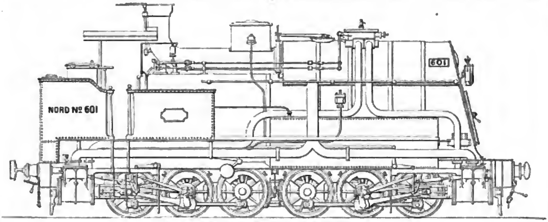
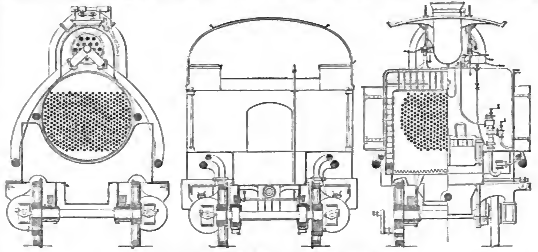


Fig. 105.—Petiet's Engine, by Gouin, Paris, 1863.

four-cylinder engines had not then, however, been actually constructed; and M. Brüll gives the diameter at 17·323 inches, equal to one pair of 24½ inches diameter. With the calculated effort of 18,747 lbs, therefore, the mean effective pressure upon the pistons throughout their stroke was 75¾ lbs. per square inch.


Figs. 106, 107, and 108.—Sections of Four-Cylinder Engine.

The two pairs of driving wheels (the second and the fifth from either end) of the four-cylinder goods engines have a lateral play on the rails of ⅞ths of an inch. Subsequently to their construction, a beam upon Baldwin's (or Beugniot's) plan was applied so as to connect the first and third axles of each group of three. This beam bears against a fixed spherical boss under the driving axle, and its ends are forked, to embrace the contiguous axles at the middle of their length, collars being provided to receive the lateral pressure of the ends of the beam. The first and third axles of each group of three have a lateral play on the rails of 1 3/16 inches, and notwithstanding the length of the wheel-base, the engines traverse curves of 262½ feet, or less than 4 chains radius. They draw trains of forty-five wagons, weighing with load 13¾ tons each, up gradients of 1 in 200, at a mean speed of 12½ miles an hour, and at an average, over the whole line, of 15 miles an hour. In this service, on the Northern Railway of France, M. Brüll reports the consumption of slack coal as 57½ lbs. per mile, or 862½ lbs. per hour. The evaporation per pound of slack coal is not given, but it can hardly be more than 8 lbs., corresponding to 110 cubic feet only per hour—a result below what might have been expected from so great an extent of heating surface.

The first four-cylinder locomotive engine made was probably that (*vide* page 21 of the present work) which Hackworth converted into the "Royal George." This did not, of course, present the conditions met with in heavy modern goods engines. In these it is necessary to distribute the great weight upon from eight to twelve points; and if the adhesion of all the wheels is to be secured, from four to six axles must either be coupled by coupling rods or more than one pair of cylinders must be employed. Although there are many examples of eight-coupled engines at work, there is a general unwillingness to couple more than three pairs of wheels, and thus a case is made out in favour of two or more pairs of cylinders. Reasoning in this manner, G. A. Nicolls, superintendent of the Reading Railroad (U. S.), designed and patented (April 25, 1848) the employment of two or more pairs of cylinders in the same engine; but he did not carry out his plan on the railway under his care. In 1854, John Cochrane exhibited a model of a four-cylinder engine at the fair of the Maryland Mechanics' Institute; but the French engineers were the first to reduce the idea to practice. The engines shown in Fig. 105 were designed by M. Petiet, engineer-in-chief of the Northern Railway of France, and they were constructed by Messrs. Gouin and Co., of Paris. A number of four-cylinder passenger engines were also made at the same time. In these the boiler, dryer, and chimney are arranged as in Figs. 105, 106, 107, and 108, and the cylinders are similarly placed at the extreme ends of the engine. The driving wheels, 5 feet 3 inches in diameter, cover a wheel-base of 17 feet, and between them are placed three pairs of smaller carrying wheels, thus making ten wheels in all. The four cylinders are each 14 3/16 inches in diameter, equal to a pair of 20 1/16 inches. The length of stroke is 13 7/16 inches only. The grate area is 20¼ square feet, the fire box heating surface is 109 square feet, and 356 tubes, 1 11/16 inches in diameter and 11 feet 5¾ inches long, present about 1,700 square feet, in addition to a dryer of 129 square feet. The whole weight of engine, with 6 tons 18 cwt. of water in the tank and 2 tons of coal in the boxes, is 47 tons, of which nearly 21 tons are on the four driving wheels, the greatest weight on a single pair being 11 tons. These engines easily attain a speed of 45 miles an hour; and the French engineers aver that the 5-feet 3-inch driving wheels are preferable, especially on account of the saving of dead weight, to those of 7 feet 6 inches and 6 feet 6 inches formerly employed in other engines for the same service.

With all that has been done, the problem of the best construction of locomotives of great power has not been solved to the general satisfaction of engineers. In most of the engines illustrated in the last few pages, from three to four pairs of wheels are coupled together; while the wheel-base is too short for steadiness in some, and too long for free working through curves in others. The objections to coupling more than two pairs of wheels are obvious enough: increased friction on the pins of the coupling rods, and, more especially, a sensible loss of power in the constant although unperceived slipping, or grinding upon the rails, of one or more of the coupled wheels, which, as a rule, never retain their exact equality of diameter for any length of time. There is always the risk, too, of an inexact adjustment of the length of the coupling rods; and if these are at all too long or too short, the consequent strain at each revolution may be very great. Where, on the other hand, the adhesion of a 40-ton or 45-ton engine is found necessary, few engineers would be willing to place from 13 tons to 15 tons upon each pair of wheels, although, with steel tyres and steel rails, and with improved permanent way, it is not settled what is the economical limit of weight upon a single point. Among others, Mr. Crampton had foreseen, as early as 1846, the importance of reconciling the various conditions involved in the problem of great locomotive traction, and his plan (patented August 25, 1846) was to couple two ordinary engines together by their foot-plates, beneath which were the coke boxes, while a tender for water occupied a position between the smoke-box end of one of the engines and the train. This system (with tank engines, however) was adopted for working the Giovi incline of the Turin and Genoa line, where it was first brought into use in December, 1853. The Giovi incline is 6 miles long, with an average rise of 1 in 36; 1½ miles, however, rising at the rate of 1 in 29. There are 2½ miles of tunnelling; and here, owing to damp, the adhesion cannot be depended upon to more than one-tenth of the weight. The tank engines originally made for this incline were by Messrs. Stephenson, and had outside 14-inch cylinders, 22-inch stroke, and four coupled wheels 3 feet 6 inches in diameter. The pair of engines weighed 55¼ tons in working trim, and took up from 70 tons to 100 tons, according to the weather, at a mean rate of 11 miles* an hour. Subsequently, pairs of six-coupled engines have been adopted, with 16-inch cylinders, 22-inch stroke, and 4-feet wheels, the pair having 2,163 square feet of heating surface, and weighing 65 tons in working trim. Owing to the want of adhesion, especially in the tunnel, the boiler pressure is fixed at 88 lbs. only in the four-coupled, and at 100 lbs. in the six-coupled engines. The latter take up ten loaded wagons, weighing 118 tons, at the mean rate of 10 miles an hour. It should be added, that the minimum radius of curvature on the Giovi incline is 20 chains.

The system of working tank engines in pairs (one driver and two firemen being employed) was adopted by the late Mr. Berkeley for working the Bhore and Thul Ghât inclines of the Great Indian Peninsula Railway. The Bhore Ghât incline is 15¾ miles long, with an average rise of 1 in 45·7. One portion, 1¼ miles long, rises at the rate of 1 in 37; and another, upwards of 5 miles long, is inclined 1 in 40. The minimum radius of curvature is 15 chains. The Thul Ghât incline is 9¼ miles long, and of this 4 miles 29 chains are on a continuous gradient of 1 in 37. The least radius of curves is 17 chains. The tank engines, originally designed to work in pairs, had 15-inch cylinders, 22-inch stroke, and four coupled wheels 4 feet in diameter. Each engine weighed 34½ tons, of which 15 tons 17 cwt. were on one axle, and 18 tons 13 cwt. on the other. The springs to carry these enormous weights had 22 plates each, and weighed 213 lbs., their deflection being ⅝ths of an inch only under a load of 10 tons. On the opening of the inclines, however, the ten-wheel tank engines shown in Plate VI. were adopted. These were designed by Mr. Kershaw, and constructed at the Atlas Works, Manchester. They have 20-inch cylinders, 24-inch stroke, and the six coupled driving wheels are 4 feet 4 inches in diameter.

At a time when the best construction of heavy goods engines is so much debated, even the unexecuted designs of engineers who have studied the subject in its broader aspects deserve notice. Among these, one of the most striking is that sent by M. Meyer, of Vienna, to the International Exhibition of 1862. Here the boiler—6 feet in diameter, and containing 348 tubes nearly 17 feet long—is mounted on two six-wheeled bogies, the pivots of which are 22 feet apart, while the wheel-base of each bogie is 8 feet 6 inches, making a total base of 30 feet 6 inches. The front bogie takes its load wholly on the centre, while the hind bogie bears its share of the boiler upon two side bearings, in the manner in which the tanks of the eight-wheeled tenders of American engines are supported. The two bogies, however, instead of being connected together by the boiler, are joined by a long coupling rod of great strength. Each bogie has a pair of 17¼-inch cylinders, with 19¾-inch stroke of piston, the wheels, 3 feet 10 inches in diameter, being coupled by rods in the ordinary manner. An ingenious system of flexible steam and exhaust pipes is provided to accommodate the varying motions of the bogies in working through curves. The middle axle of each bogie has a considerable allowance of end play, so that, notwithstanding the great total length upon the rails, the engine could traverse curves of small radius. The whole weight is 60 tons, giving 10 tons to each pair of wheels. As bearing upon the flexibility of M. Meyer's arrangement, it may be observed, that the passenger carriages of the American railways have two bogies upwards of 25 feet, and in some cases more than 30 feet, from centre to centre of pivots, the wheel-base of each bogie being from 4 feet 6 inches to 6 feet, and that these run freely through curves of 500 feet radius, while they are sometimes drawn by horses through curves of less than 100 feet radius around the street corners of towns. M. Meyer's estimate of the tractive force of his engine is 22,000 lbs., equal to the draught of 2,300 tons, exclusive of engine, on a level, 340 tons up 1 in 40, or 155 tons up 1 in 17. An effort of 22,000 lbs. at 880 feet per minute (10 miles an hour) corresponds to 587 horse-power. This estimate does not appear unreasonable; and, indeed, under the vigorous draught in the chimney of a *passenger* engine, a boiler of such great size should afford steam equivalent to at least 1,500 horse-power. It is not to be overlooked that such great tractive force would necessitate the construction of wagons of much more than the usual strength of frame, as the ordinary rolling stock of English lines would be quickly broken up under a traction of 10 tons, aggravated by the frequently arrested momentum of trains of many hundreds of tons weight.*

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* Mr. Drysdale has given the rate of speed as 15 miles an hour, but the late Mr. Wealleus (formerly of Messrs. Stephenson's) has corrected this.

In pursuing the course of recent innovation with respect to goods engines of great power, we have for the time neglected many other points in which the locomotive of the present day differs from that of but a few years ago. The introduction, in 1853, of what was substantially the American class of steep-gradient engine upon the Semmering incline, has led, as has been shown, to a series of alterations, striking in respect of appearance if not in principle, and it has been deemed better to trace these in their own order and by themselves. In the case of Engerth's, Beugniot's, Petiet's, and Meyer's engines, there is probably nothing which could not have been foreseen, after an easy induction, by any engineer not bound by prescription and accustomed usage, and it is still a moot point whether much has been gained upon the former practice with lighter engines and lighter trains. It is the fact that the old six-coupled engine still holds its ground on English lines, and it will doubtless continue to do so until further and more accurate knowledge is obtained as to the cost of working and repairing the remarkable examples which illustrate the ingenuity of our professional brethren on the Continent.

Turning from the subject of steep-gradient engines, the progress which has been made in locomotive practice within the last dozen years is most apparent in the substitution of coal in place of coke and wood. The inside-cylinder arrangement, too, which has been to a great extent abandoned on the Continent, and wholly so in the American practice, has long since ceased to receive the almost exclusive favour which it once had in England. A general increase has taken place in the dimensions and weight of locomotives, not that absolutely heavier passenger engines are now in use than could be found, here and there, in 1852, but that the number of heavy engines has very greatly increased. Attention has been notably directed, also, to the adaptation of locomotives to sharply curved lines. The employment of steel axles and tyres, the Injector, Mr. Ramsbottom's self-filling tender and his self-packing piston and gravity lubricator, are recent, although minor improvements. Feed-water heaters, in various forms, were in use many years ago, and new applications have appeared more recently, but the partiality for the Injector (which cannot act when the feed-water is much hotter than 100°) has checked their further introduction.

In respect of burning raw coal in locomotives, Gray and Chanter's early experiments upon the Liverpool and Manchester line having proved fruitless, the first encouraging results in this direction appear to have been obtained by the late Mr. Samuel Hall in 1841. He then applied his patented plan to the engine "Bee," on the Midland Railway. Sixteen tubes in the lower three rows of the boiler were prolonged, by means of movable pipes, through the smoke-box door, where they were open, and thus received and directed as many streams of air into the fire box, just over the level of the burning coal. In each of the two sides of the fire box also, four short tubes or hollow stays passed through the water space, thus making in all 24 air openings, 1¼ inches each in diameter, just above the burning fuel. A steam jet or blower was provided in the chimney, so as to keep down smoke when standing. According to the report of the engineer of the North Midland line, Mr. Marshall, the plan answered well, giving a good supply of steam, with little or no smoke. Mr. Charles Wye Williams had, as early as 1840, written his well-known work, *The Combustion of Coal;* and in 1839 he had patented his system of admitting a great number of fine streams of air over the fire. His ideas were in part adopted by the late Mr. Dewrance, who, in 1845, altered the engine "Condor" to burn coal. Mr. Dewrance divided the fire box by a transverse water space, traversed by short tubes, and he kept the fire in the back division, while that next to the barrel of the boiler served as a combustion chamber, its bottom being closed by a plate, through which opened an air pipe, with a head perforated like the rose of a watering pot. This arrangement is understood to have answered tolerably well, but it was not repeated in other engines.

After Mr. M'Connell, whose engines have been mentioned on page 73, Mr. Joseph Beattie, of the London and South-Western Railway, was the first to burn coal, without smoke, in the regular working of a great passenger traffic; and he appears to have been the very first to obtain the full evaporative duty of coal in locomotives. In 1853, Mr. Beattie altered the boiler of an ordinary engine, so as to burn a proportion of coal with the coke; and early in the following year the first of a new class of engines, with his further improvements for burning coal only, commenced working. The alteration of 1853 consisted in the addition of a small supplementary fire box and fire grate to the existing boiler. Coke was burnt in the original fire box, but a mixture of coal and coke was fed to the supplementary box, from which the products of combustion passed through a series of short tubes into the space over the coke fire, and thence into the boiler tubes as usual.* In the boilers designed for the use of coal only, the fire box was divided by an inclined water partition into two compartments, each having its own door, fire grate, ash pan, and damper, as shown in Fig. 109. The principal fire was maintained in the box nearest to the foot-plate. The gases rising from the coal were met by a great number of fine streams of air entering through the perforated door, and both the gas and air rose through a grating of fire-clay tiles into the upper part of the second fire box (on the grate of which coal was burnt only slowly, with a slight and carefully regulated admission of air through the front damper). The mingling air and gases were further deflected downwards by a hanging water bridge, and over a fire-brick arch, and through a series of fire-clay tubes, into a combustion chamber, from which the boiler tubes led into the smoke box. The whole arrangement can be traced in Fig. 109, and also in Plate XVI. Mr. Beattie had very carefully provided—1st, for a sufficient admission of air in fine streams over the fire; 2d, for the intimate mixture of the air and gases; and 3d, for the maintenance of a temperature throughout the combustion chamber sufficient to ignite this mixture at whatever point it was ready to burst into flame. The front fire box and the nearly incandescent fire bricks supplied an intensity of heat which, in the presence of the saturated gas, served the purpose of a spark to a train of gunpowder. Without the fire bricks, the boiler shown in Fig. 109 could not be fired with coal without producing smoke. Mr. C. Wye Williams has for a long time, and very justly, urged that the intimate, or, more strictly, the atomic mixture of coal gas and air, as the former rises from the fire, is sufficient to insure good combustion; but under the rapid draught, and with the short "run" in locomotive furnaces, it appears certain that this mixture cannot be effected so rapidly that the progressive inflammation from particle to particle of the gaseous volume in course of saturation can be depended upon to complete the full chemical change requisite in order to liberate all the heat and to prevent smoke.

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* In 1863, Mr. Millholland, of the Reading Railroad, U.S., completed and put to work an engine of the following dimensions, and for the purpose of assisting heavy coal trains up a gradient of 1 in 135, and 1¼ miles long:—Outside horizontal cylinders, 20 inches in diameter, with 26-inch stroke of piston. Twelve coupled wheels (six pairs) 3 feet 7 inches in diameter, each pair being 3 feet 11 inches between centres, giving a wheel-base of 19 feet 7 inches. The fire grate is formed of water tubes, through which a constant circulation is maintained. It is 9 feet long and 3½ feet wide, presenting 31½ square feet, and is fed with anthracite coal. The boiler is 48 inches in diameter, and besides a combustion chamber 2 feet 1 inch long, it contains 174 iron tubes 2 inches in diameter and 13 feet 6 inches long. The total heating surface is 1,428 square feet. There are three water tanks, containing 1,224 American, or say 1,000 imperial, gallons. No coal is carried, except upon the fire bars, the fire being replenished at the ends of the journeys. The steam ports are 18¾ inches long and 1⅝ inches wide, with ⅛-inch lap on the valves. The exhaust port is 3¼ inches wide. The blast orifice can be varied between 11 square inches and 24 square inches. The connecting rods are 11 feet 4 inches long. The engine is 36 feet long and 8 feet 6 inches wide, and its weight is 100,320 lbs., or 44¾ tons. With a mean effective pressure of 70 lbs. per square inch on the pistons, and without allowing for the friction of the engine, it would exert a tractive force of 16,930 lbs., equal to the draught, at a slow speed, of upwards of 2,500 tons on a level, the resistance to traction being often as low as 6 lbs. per ton.

* Drawings and descriptions of the various coal-burning locomotive

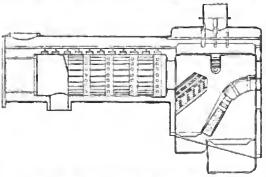

Fig. 109.—Beattie's Coal-Burning Boiler, 1854 to 1859.

Mr. Beattie's coal-burning engines are in general use on the London and South-Western Railway, and a considerable number of them are at work on other lines. But the boilers are heavy and costly; and on this account they have not been as widely adopted as, for the comfort of railway passengers, might be wished. When it was attempted, ten years ago, to burn coal in locomotives, the prevention of smoke was generally reckoned an indispensable condition of success. In their anxiety, however, to avail themselves of the cheapest fuel, with the least alteration of the locomotive, engineers have presumed more and more upon public tolerance, until, at last, coal has almost everywhere taken the place of coke on railways, and that with but few, and these often the crudest, special means for its combustion; and, as a consequence of this, in many cases, the atmosphere of the train has become a sooty and acrid pollution with which the clauses of the Smoke Prevention Act are, unluckily, powerless to deal. When it was found that coal could be made to answer, it was next sought to burn "slack." This, from its offering a greater resistance to the passage of the air, required a larger grate, and it is thus that such ample fire boxes are to be found in the recent French engines, in which slack coal is the principal fuel. In 1857, Mr. Cudworth, of the South-Eastern Railway, adopted the long, inclined double grate shown in Plates V. and XI.; and with alternate firing through each door, and with the opportunity for the mixing and ignition of the gas and air in front of the longitudinal midfeather, this is an effective and favourite form of fire box. The bars are less likely to clinker than when placed horizontally, and they can be much more easily cleaned when this becomes necessary.

The practice of admitting air into the fire box, above the level of the burning fuel, is now followed wherever coal is burnt in locomotives. When it is taken in through the door, the air commonly reaches the tubes before any effective mixture with the rising gases can possibly take place. Whatever variety, therefore, is to be found in the boilers of coal-burning locomotives appears chiefly in the contrivances for mixing the gas and air while they are momentarily in each other's presence in the fire box. In some cases the door is either left partly open or altogether removed, and a shovel-shaped iron plate is so placed in the door-hole as to throw the entering air directly down upon the fire, the plate being lifted out of the way whenever fresh coals are added. This is the readiest make-shift for the purpose, and, as far as it goes, it embodies a sound principle. Mr. Hall's air tubes and hollow stay bolts, already described, are still employed in some cases, and they are rather more effective means to the same end. In some cases jets of live steam have been directed, as proposed by Nicholson in 1806, through air openings over the fuel. In a locomotive engine, however, it is evident that the air opening once made, enough air will enter in virtue of the chimney draught, without the need of other means of forcing. The steam jets are not necessary for "inducing" the entrance of air, however effective they may be in mixing it with the gases in the fire box. The effect, however, in this respect, has been found to be altogether disproportionate to the expenditure of steam; and in most cases, therefore, where the steam jets have been tried, they have been abandoned.

To promote the mixture of air with the gases rising from the fuel, it is better to introduce the air in a direction opposite to that of the current through the tubes.

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boilers in use between 1853 and 1858 may be found in *European Railways* (New York, 1858), and in *Recent Practice in the Locomotive Engine* (Glasgow, 1860), of both of which works Mr. Colburn is a joint author.

The air admitted over the fire will thus be compelled to change its course while passing through the fire box, and this change is favourable to intimate mixture. In Mr. Ramsbottom's engine, Plate II., two air openings of good size, and regulated by dampers, are made through the front or tube-plate wall of the fire box, and a few inches above the surface of the coal on the bars. The air entering through these holes is deflected downward and along the surface of the bed of fuel by means of an arch of fire bricks, built against and projecting for some distance from the tube plate. By the time the air turns the edge of this arch and rises towards the tubes, a tolerably thorough mixture with the rising gases will have been effected. At all events, with care in firing—and this is the one great desideratum in all coal burning—the arrangement just described answers its purpose remarkably well.

Various as have been the plans for burning coal in English locomotives, as great a number and a much more remarkable description of contrivances for the same purpose have appeared in America. Dimpfel's water-tube boiler,* Fig. 110, is in use for express and other engines on the Philadelphia, Wilmington, and Baltimore Railroad. The tubes—of iron and of small diameter—open through the crown plate of the fire box, and, bending below it, extend to and communicate with a water space at the smoke-box end of the boiler. Although thus traversed by tubes, the whole length of the barrel of the boiler serves as a combustion chamber, the outlet for the products of combustion being at the bottom of the smoke box. The circulation through the tubes is vigorous, not to say violent—sufficient, indeed, as has been ascertained, to bring a half-pound bolt, purposely left in one of them, up to the crown plate. The boiler is heavy, however, in proportion to its heating surface; it requires much staying; it is not easy to identify a leaky tube whenever leaking is known to be going on; and while the removal of any tube is difficult, a considerable number often require to be taken out in order to properly get access to but one.

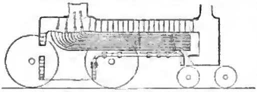

Fig. 110.—Dimpfel's Water-Tube Coal-Burning Boiler, 1849.

Boardman's boiler,† Fig. 111, was at one time in considerable use in the States. A large ⌂-shaped flue extends from the fire box directly through to the smoke box, a plate iron partition being fixed quite across the flue at about two-thirds of its length from the fire box. A series of vertical tubes descend from the flat bottom of this flue to an ash pan below; but, by the interposition of the partition just mentioned, the heated currents go down through but about two-thirds of the total number of tubes, and rise again through the remainder, escaping thence (after having given off most of their heat) into the smoke box. An air pipe, perforated at its discharging end, is placed in the main flue to promote combustion. This boiler is heavy, particularly on the leading wheels; its flat sides require very extensive staying, and the restoration of a single stay rod, owing to the difficulty of access, has cost as much as £120.

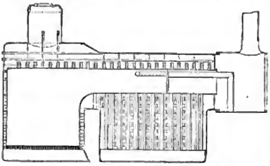

Fig. 111.—Boardman's Coal-Burning Boiler, 1853.

Phleger's boiler, Fig. 112, is notable merely for its hanging bridge and "water wall," and for its fire grate, formed of water tubes. Messrs. Norris made boilers at one time upon this plan, and of great size. For a ten-wheel goods engine, with six coupled wheels 4 feet in diameter, and having 17-inch cylinders and 24-inch stroke, the boiler was 24 feet long over all, the fire box and combustion chamber being 11 feet long, while the tubes—230 in number and 2 inches in diameter—were also 11 feet long. The barrel of the boiler, 3 feet 10½ inches in diameter, was filled with tubes, and a large steam chamber was formed above it.

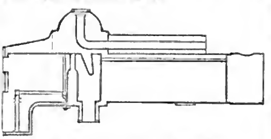

Fig. 112.—Phleger's Coal-Burning Boiler, 1852.

Passing from coal-burning boilers to outside-cylinder engines, it may be said that the type which is now meeting with so much favour on English lines is a growth from the practice of the late Mr. Locke. He was engaged under George Stephenson upon the Liverpool and Manchester Railway at the time when the outside-cylinder engines of the "Northumbrian" class were discarded; but this did not prevent him from adopting an improved form of these engines on the Grand Junction, and subsequently on the London and South-Western, Western of France, and Caledonian Railways. The "Crewe" engine of twenty-five years ago has grown into the "Lady of the Lake" pattern of 1862, Plates I. and II. The engines which Mr. Locke employed Mr. Buddicom to make at Rouen have been the parents of outside-cylinder engines in many parts of the Continent; while the Caledonian and the South-Western engines have taken the forms shown in Plates VII. and VIII., and Plates XVI. and XVII.; and Mr. Sinclair, formerly of the Caledonian

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* English Patent to W. E. Newton, April 17, 1849.

† English Patent to C. D. Garlissal, March 29, 1856.

Railway, preserves the family likeness in his present practice on the Great Eastern Railway, as will be seen in Plates IX. and X. The outside-cylinder engines have been improved by placing the cylinders in a nearly or quite horizontal position, by strengthening the cylinder fastenings, and by guarding more carefully against radiation, and, finally, by counterweighting a portion of the mass of the reciprocating parts; for few, we believe, counterweight their whole weight. With these improvements, the engines illustrated in the plates just cited run as steadily, and are as economical of fuel, as any inside-cylinder engines in use; while, also, the former are simpler, lighter, and proportionately less top-heavy. And the frequent breaking of crank axles in inside-cylinder engines is of course prevented where there is no crank axle to break. There is yet room for improvement in the fastening of outside cylinders to the framing; for there are but few of this description of engine in which the cylinders do not, after a short time, work loose to some extent, when the pistons are under a full pressure of steam. The Americans, who now make outside-cylinder engines exclusively, pay much attention to this point; and however we may criticise their practice, it is most effective in the respect under consideration.

The introduction of steel axles, and especially steel tyres, has been attended with important advantages in respect both of economy and safety. It is only within a few years that the art of producing cast steel in masses of any considerable weight has been known. The fear that steel, when subjected to the shocks of railway work, would break, has been dispelled by the production of qualities far stronger and less brittle than wrought iron. For crank axles, indeed, steel is the only material that will not break. Its durability, when applied to the trying service of engine tyres, has passed into a proverb; and it is found, moreover, that, so far as adhesion is concerned, steel is in no degree inferior to wrought iron in its bite upon the rails. For boiler plates and for tubes steel has not given the same degree of satisfaction, although there are steel boilers in use which have stood well for a number of years. The term "steel" is now applied to materials of very various qualities, and it may be that the particular grade suited to boiler work will yet be identified, and produced with certainty. Some of the brands now made will not weld; others are burnt at very moderate heats; and a good deal of plate steel has been found brittle under the punch. So great, however, has been the progress lately made in the steel trade, that the correction of these evils is not perhaps impossible.

The earlier engines always stood upon a short wheel-base, generally 5 feet for those with four wheels, and from 10 feet to 12 feet for the six-wheel classes. The leading engineers, too, spared no expense in order to obtain lines as nearly straight as possible. In the main line between London and Birmingham there is no curve with a radius less than 1,800 feet; while the Great Western minimum is at least twice this. But within the last few years, and while the wheel-base of heavy passenger engines was being lengthened to 16 feet, and in some cases to 18 feet, curves of 20 chains, and here and there of even less than 10 chains' radius, have been introduced on the newer lines. Now, the versed sine of an arc of 600 feet radius, and the chord of which is 16 feet, is $\frac{7}{11}$ths of an inch nearly; the inner rail is 1 foot shorter than the outer in every 132 feet run around the curve; and with parallel axles the extreme wheels on the inner rail are $1\frac{1}{8}$ inches further apart than they would be if the axles took each a position radial to the curve. At 30 miles an hour, corresponding to 44 feet per second, the centrifugal force of a train on a curve of 10 chains' radius, or one-fourth mile in diameter, is* 0·0912 of its own weight. So far as this force is concerned, it can be neutralized by raising the outer rail of the curve until the angle of the inclination across the line is such that 0·0912 of the weight of the train would, if it were at rest, be supported against the inner rail. In other words, taking the gauge of the line as 1, the outer rail should be 0·0912 higher than the inner rail on the curve of 660 feet radius, and for trains at 30 miles an hour. For the ordinary gauge of 4 feet 8½ inches, or say 4 feet 11 inches, from centre to centre of the bearings of the wheels on the rails, this is 5⅜ inches for the super elevation of the outer rail. But even then, supposing the tyres of the wheels to be cylindrical, or without "cone," there is a loss of power, in the slipping of the inner wheel to an extent equal to its own circumference in every 132 revolutions. For every 132 feet run by the wheel on the outer rail of a 10-chain curve, the inner wheel must slip 1 foot; and at 30 miles an hour, or 2,640 feet per minute, this slip will be 20 feet per minute. If the whole weight of engine and train be 100 tons, 50 tons will rest on the inner rail, and the adhesion or friction of this weight may, with a clean dry rail, amount to one-fourth (or more) of this—say 12½ tons. Now, to overcome a friction of 12½ tons, or 28,000 lbs., at the rate of 20 feet per minute, requires $\frac{28{,}000 \times 20}{33{,}000} = 17$ horse-power. There is still another loss of power, to wit: with exactly parallel axles, the wheels, if cylindrical and of the same diameter, cannot move in a direction parallel to the curve of the line, otherwise than as they are forced by the pressure between the flanges and the rails. Irrespective of centrifugal force, they tend to advance in a direction which would carry them across rather than along the line, exactly as if a carriage with axles not parallel to each other were run upon a straight line. In the case already assumed, the end axles make an angle, with a tangent to the curve, corresponding to a lateral deviation of 1 in 84. In other words, for every 84 feet run upon a 10-chain curve, the leading wheels of an ordinary engine with 16-feet centres must move 1 foot laterally upon the rails; and this quite irrespective of centrifugal force, and of the slipping of the wheels upon the inner rail. At 30 miles per hour this lateral sliding would be 31½ feet nearly per minute; and the force required to effect it may be inferred from the fact that, before the adoption of steel tyres, the flanges of the leading wheels of the engines running through one curve of 5 chains' radius, on the North Woolwich line, were

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[*] The centrifugal force of a revolving body is found by the formula, $\frac{\left(\frac{v}{4\cdot01}\right)^2}{D}$ C =, where v = velocity of motion in feet per second, D the diameter, in feet, of the circle of revolution, and C the centrifugal force, the weight being 1.

regularly cut away, so as to require to be newly turned every six weeks.

This partial statement of the nature of the resistances due to curvature will serve not only to introduce a notice of the radial running gear now in course of application to the English locomotive, but also to point the history of the various inventions by which it has been sought to drive railway rolling stock around a corner with the least loss of power. In 1812, William Chapman, a civil engineer of considerable eminence in the North, obtained a patent (No. 3,632), in the specification of which, among other things, he described the four-wheel swivelling truck, or, as the pitmen near the Border nicknamed it, the "bogie"—a name which a reformed technical nomenclature may be expected to supersede with a worthier appellation. Chapman proposed to draw coal trains along a railway by gearing the engine wheels, not into a rack, as Blenkinsop had at the time just begun to do, but into a chain laid down along the line;[*] and it was to enable the engine or driving carriage to pass through short curves that he designed the swivelling truck. In ordinary cases six wheels were to be employed, the axle of one pair being held at right angles to the frame; but for heavier and longer engines two four-wheel trucks were recommended. Chapman's drawing is meagre, but his description is clear; and the truck was undoubtedly his invention. His chain gearing was employed for a short time on the railway of the Heaton Colliery; but whether his truck was also in use there is not recorded. The principle of the truck is not only simple enough, but it is contained in the locking wheels and perch-bolt of the earliest four-wheel carriages, and it was embodied in the locomotive of Cugnot, and in the model by Murdoch, both in the last century. In these, however, the leading wheels had to be previously turned, in order to pass around a curve; but in the railway carriage the act of turning adjusts the truck.

It has been asserted that Hedley's eight-wheel engine (Fig. 14 of the present work) had two swivelling trucks. The author, however, has found no evidence in support of this statement; and, moreover, the drawing contained in the first edition of Wood *On Railroads* appears to show that the four axles were necessarily always parallel with each other. The eight wheels were employed to distribute the weight of the engines over an increased number of points upon a light plate rail. This was replaced by an edge rail in 1830, and the engines were then altered to work upon four wheels.

It has been represented, also, that the eight-wheel wagon described and illustrated in Tredgold *On Railroads* (1825) had swivelling trucks; but there is nothing in the description, nor, indeed, in Tredgold's sketch, to warrant this view. His object was to distribute the weight so that it should be more nearly equal upon all the wheels when passing over irregularities in the line. The same object is more fully stated, and means of effecting it are described (and which are probably identical with those intended by Tredgold), in the specification of a patent of William Chapman's, No. 5,540, A. D. 1827. Here, although no springs were employed, "equalizing bars" are described, and their purpose explained in terms which would apply also to Tredgold's sketch.

The truck, nevertheless, became well known after the date of Chapman's patent of 1812, and it was employed in a rude form on the Newcastle quays. The late Mr. Robert Stephenson once informed the author that when, in 1828, a commission of American engineers—Messrs. Whistler, Knight, and Mac Neill—visited Newcastle, he recommended them to adopt the truck for the engines which they contemplated working through curves of 400 feet radius upon the Baltimore and Ohio Railroad, and other lines then projected in the States.[*]

Besides the truck, there was another mode of adapting railway wheels to curves. This was by making the tyres conical instead of cylindrical on their wearing faces, the larger diameter of the conical frustrum being next the flange. Thus, when the outer wheel of a pair was forced centrifugally against the outer rail, it acquired a larger, and the inner wheel a smaller, effective diameter, thereby adapting the pair of wheels to a curve of which the radii of the inner and outer rails respectively are to each other in the same proportion as the smaller and larger diameters on which the pair of wheels run. Thus, if one wheel be running on a diameter of 3 feet and the other of 3 feet 1·2 inches (3·1 feet), and the gauge be 5 feet (to be measured, in this case, between the bearing points of the wheels on the rails), then 0·1 foot : 3 feet :: 5 feet : 150 feet radius of inner rail of corresponding curve, through which the pair of wheels will roll without sliding. Railway wheels were slightly coned, and, indeed, the tread was united to the flange by an easy curve, at a very early period, but with the object, not of passing freely through curves, but of preventing the otherwise almost constant rubbing of one or the other flange against the rail. This was the only object recognized by Tredgold,[†] in 1825, and by Wood,[‡] as late as 1832, as that with which railway wheels were coned. In 1825, however, William T. James patented a railway wheel, with its tread formed in steps of different diameters, and the cast-iron rails were to be so formed and laid that on curves the smaller diameter of the inner wheel and the larger diameter of the outer wheel should bear simultaneously on the corresponding portions of their respective rails.[§] The same thing was subsequently patented in the States by Joseph Wright,[‖] and again, afterwards, in France, by M. Laignel.[**] Although never employed in the precise form proposed by these several patentees, the idea was turned to good account in the States in the following manner:—In many cases there, railways are extended through the streets of the larger towns and cities, the trains being drawn by horses. In turning from one street into another at right angles, curves of from 60 feet to 80 feet radius only are often inevitable. In order to cause the least interference with the ordinary street traffic, the rails are flat bars, with a longitudinal groove to receive the flanges of the wheels. But on the short curves the outer rail has no groove, and the flange, instead of the tread, of the wheel runs upon it. The ordinary railway wheels are 2 feet 6 inches in diameter, or 2 feet 8½ inches over the flanges; and thus they work around curves of 60 feet radius without slipping. M. Laignel had proposed to work through curves of 33 feet radius in the same general manner.

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[*] Hedley, to whom so much credit has been given for the early employment of the adhesion between smooth-driving wheels and smooth rails, proposed an imperfect application of Chapman's plan in his patent of 1814. *Vide* p. 17 of the present work.

[*] The charter, from the State of Maryland, of the Baltimore and Ohio Railroad Company is dated February 28, 1827.

[†] Tredgold *On Railroads*, p. 43, and Fig. 24, Plate III.

[‡] Wood *On Railroads*, second edition, p. 67.

[§] W. T. James' patent, March 5, 1825, No. 5,117.

[‖] Joseph Wright's patent, April 17, 1829. Vide *Franklin Institute Journal* for 1829, second series, vol. iv., p. 270.

[**] Laignel's patent, July 23, 1830. Vide *Brevets d'Invention*, vol. lviii., p. 145, and Plate XII.; *vide*, also, *Bulletin de la Société d'Encouragement* for 1843, p. 352.

Ross Winans appears to have been the first[*] to point out the *rationale* of coned wheels on railways. Although there was much in his specification that was impracticable, still it dwelt upon the importance of both axles of a wagon taking positions radial to the curves through which they passed; and it contained, moreover, the first proposition, so far as the author can learn, to employ outside instead of inside journals to the axles. Soon afterwards Joseph Wright[†] obtained a further American patent for coned wheels and radial axles, the action of which he described. Although the date of this patent is more than three months after that of Winans, the specification of the former was drawn, under the American law, sometime previous to its issue; while that of the latter was filed in November, 1829, by which time Wright's specification had been printed in the *Journal of the Franklin Institute.* In M. Laignel's French patent of 1830, already referred to, the action of coned wheels was also described.

It was long before the swivelling truck was introduced into English locomotive practice. The engines made, about 1846, for the South Devon line, were probably the first to be fitted with it. Coned wheels, however, were very generally adopted, the cone amounting to one-seventh of the width of the working face of the tyre.[‡] But with parallel axles, the driving wheels being near the middle of the wheel-base, are forced, not against the outer, but towards the *inner* rail, in passing through curves, the engine occupying the position of the chord of an arc. The cone often given to the tyres of the driving wheels was thus in the wrong direction, the smaller diameter being made to roll on the outer, and the larger diameter on the inner, rail. Cylindrical tyres would have answered a better purpose, it being understood that these are to be preferred, not for the leading and trailing wheels, but for wheels at or near the mid-length of the wheel-base of engines of which all the axles are parallel. So considerable is the movement of the driving wheels in such cases towards the inside of the curve, that the flange of the inner wheel is often forced violently against the rail. To this strain the frequent breaking of crank axles was formerly attributed; and the late Mr. Robert Stephenson, in his six-wheel engine patent,[§] based his claim upon the making of the driving wheels with plain tyres, or tyres without flanges. These were extensively used for a time, but of late years flanged tyres have been employed for all the wheels of the engine.

The swivelling truck was first applied to locomotives in the States in 1832, and, with the exception of engines having all their wheels coupled, none are now made there without it. It had been applied to passenger carriages as early as 1831, but until 1835 the draught was directly upon the truck, instead of through the body of the carriage.[*] Until this change was effected, the axles in each truck were placed as near each other as was permitted by the flanges of the wheels; but when the truck was no longer steadied by the pull upon the couplings, the axles were placed further apart, until at last their centres have been fixed at a distance from each other equal to, and sometimes greater than, that of the gauge of the line, this being the least distance at which steadiness is effectually secured at high speeds.[†]

In 1837, Mr. William Bridges Adams, in his work, *English Pleasure Carriages*, proposed a system of two-wheel railway vehicles, so coupled as to permit of the greatest flexibility in passing through short curves. It is evident, however, that, without special couplings of great strength, such carriages would have a decided vertical unsteadiness; nor could the equal loading essential to keeping them in balance be depended upon.

In 1838, M. Claude Arnoux[‡] obtained a French patent for a system of rolling stock which he, as President of the Paris and Orsay Railway Company, introduced upon that line (then extending only to Sceaux) in 1846. At the Paris, Orsay, and Sceaux stations, instead of turntables, the line makes a nearly complete circle of 82 feet radius; and between Bourg-la-Reine and Sceaux there are five semicircles of small diameter. Upon these trains have worked for the last eighteen years, at the rate of 20 miles an hour. The wheels of the carriages are loose upon their axles, and by a vertebrated arrangement of the body framing, each axle takes a radial position to the curve through which it may be passing. A series of guide wheels is also placed obliquely, so as to bear, on occasion, against the inner edge of the head or table of the rail. This system of running gear was afterwards patented in England.[§]

In the States the axles of the truck wheels were not only spread from their original distance of 3 feet between centres to 4 feet 6 inches, 5 feet, and in some cases 6 feet, but six-wheel trucks were occasionally made. Such were the trucks of the Crampton engines made by Norris in 1849, and the Michigan Central, Camden and Amboy, Illinois Central, Great Western of Canada, Grand Trunk, and other railroad companies, successively adopted six-wheel trucks for their passenger carriages. As applied to passenger stock, the American truck is always provided with a peculiar arrangement, permitting of easy lateral motion of the carriage upon its supports. This is known as the "swing-beam," patented by Davenport and Bridges, May 4, 1841. The pivot is formed and the load supported, not directly upon the frame of the truck, but upon a bolster which is hung from the truck frame by means of long links, free to swing in a direction at right angles to the length of the carriage. With the swing-beam the carriages run with greater ease and steadiness, not only around curves, but upon rough portions of the line. More recently the swing-beam has been applied to engine trucks, and with good results. For engines the practice was adopted as early as 1837 of supporting the whole load of the truck upon its centre, thus dispensing with side bearings; and this practice is now followed in respect of the trucks of all American engines, and also for the front truck of the tender, which is thus carried on three points. The centre bearing is not the ball and socket once employed, and still retained in certain English engines, but it is formed upon a flat, annular seat of about 12 inches diameter, and with this there is no rocking motion.

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[*] Ross Winans' English patent, May 30, 1829, No. 5,796.

[†] Joseph Wright's patent, September 10, 1829. Vide *Journal of the Franklin Institute* for 1829, second series, vol. iv., p. 272.

[‡] The width of the working face of the tyre being 3½ inches, the diameter next the flange was very commonly made 1 inch larger than at the outer edge.

[§] Robert Stephenson's patent, October 7, 1833, No. 6,484.

[*] This change was made by Jacob Rupp, a foreman acting for the Baltimore and Ohio Railroad Company. The connection of the draw-bar directly with the pivot or king bolt of the truck had been practised in 1833 on the Newcastle and Frenchtown Railroad; but Rupp's plan was found preferable to this.

[†] On the 1st October, 1834, Ross Winans obtained an American patent, afterwards extended to 1855, which led to a series of remarkable legal contests upon the construction of the claim. It was asserted that it covered the plan of eight-wheel double-truck carriages in general use in the States; but this claim was not sustained. The only novel feature in Winans' patent was that of connecting the axle boxes of two pairs of wheels together by means of two strong springs, one on each side of the truck, these springs being themselves connected together by a transverse bolster carrying the swivelling pivot socket. No other framing was employed. This truck was once extensively used; but it was unsteady and unsafe at high speeds, and has been abandoned.

[‡] Arnoux's patent, March 20, 1838. Vide *Brevets d'Invention*, vol. lxxx., p. 209, Plate XVII.

[§] Jerome Deville's patent, November 3, 1838, No. 7,872.

A number of sixteen-wheel passenger carriages are also in use on some of the American lines. These are supported under each end by a swivelling frame, each end of which, in its turn, is borne by an ordinary four-wheel truck. A passenger carriage of this kind was made at Springfield, Massachusetts, in 1859, for the late Viceroy of Egypt.

In May, 1857, a plan—which, as represented, was impracticable—was illustrated in the *Artizan* journal, and proposed as a means of enabling the leading wheels of engines to assume a radial position to the curves through which they passed. It was a plan intended to give the motion accomplished by Mr. Adams' radial axle boxes, as shown in Plate XXIII., and, except in one most important respect, it was that of Mr. Adams. But the *Artizan* proposition contemplated straight instead of curved guiding surfaces; and hence, from an oversight, it was of no value, unless as a suggestion to others. In June, 1857, a French engineer, M. Edmond Roy, obtained an English patent for radial axle *brasses* (not boxes); but his invention, as tried and abandoned on one of the French lines, was defective in point of detail. Mr. Adams' plan for radiating the leading and trailing axles in curves was patented in 1863, and is illustrated in the plate just referred to. It enables engines of the usual length of wheel-base to go through curves of 2 chains' radius, and the radiating axles appear to run steadily, at high speeds, on straight as well as curved lines.

As early, at least, as 1853 the necessity for adapting the unyielding rectangular wheel-base of heavy engines to sharper and sharper curves began to be more widely recognized. The truck was then adopted for the large passenger engines of the Bristol and Exeter Railway, and, subsequently, for the engines of the North London and the Great North of Scotland lines; while it was also applied, by English and Scottish makers, to engines for the Nova-Scotian, Chilian, and Great Indian Peninsula Railways. In 1857, Mr. Bissell,* an American, made an important improvement in the truck. In its original form only the truck axles could take a practically radial position in a curve. So far as the driving and trailing wheels were concerned, they took nearly the same position in a curve which they would have if an ordinary single leading axle occupied the place of the pivot of the truck. Yet, in the States, where the truck is in general use, there are, or lately were, goods engines on the 6-feet gauge of the Erie Railway, which measure 21 feet between the centre of the truck pivot and the trailing axle; and these engines are regularly worked through curves of 955 feet, or 14½ chains' radius. In the Bissell truck the load is taken upon the centre, as in the swivelling trucks in common use in American engines previous to 1857; but the angular motion, instead of taking place around a pivot between the truck axles, is defined by a radial framing working around a stud or king-bolt at some distance behind the truck, and not far from the driving axle (See Figs. 113 and 114). Thus, in passing around a curve, the framing of the truck sways bodily to the right or left, as the case may be, under the front end of the engine, and although the equal distribution of the weight upon the truck wheels is thereby affected to some extent—the wheels on the outer side of the curve having sometimes to bear nearly one-eighth more than their normal load—this is not attended with any sensible disadvantage; while, on the other hand, the body of the engine being no longer confined to an adjustment imposed by the truck wheels, the driving and trailing axles are left free to take positions nearly radial to the curve through which the engine is moving. The lateral sliding movement between the front end of the engine and the framing of the truck, when entering upon or leaving a curve in the line, takes place, not upon a horizontal surface, but up an incline in either direction, as shown in Fig. 115. On straight lines the double-inclined bearing surfaces serve

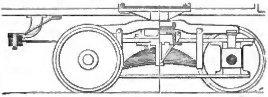
Fig. 113.—Bissell Truck, Elevation and Longitudinal Section.

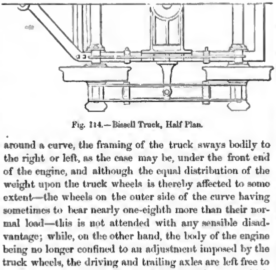
Fig. 114.—Bissell Truck, Half Plan.

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* Levi Bissell's English patent, May 5, 1857, No. 1,273.

to keep the truck steady, besides insuring an equal distribution of the load upon the truck wheels. The four-wheel Bissell truck has come into general use on the American lines, the form and details adopted being fully shown in Plate XX.

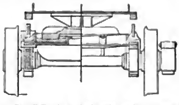

Fig. 115.—Bissell Truck, End Elevation and Transverse Section.

In September, 1857, the author recommended Mr. Bissell to apply his radial gear to a single pair of wheels, with a view to its application to English engines. The arrangement then suggested* has since been adopted upon the Great Eastern and Metropolitan lines, and, with the substitution of horizontal instead of double-inclined sliding surfaces, by some of the continental railway companies. Various modifications of the Bissell truck have been made within the last two or three years. In one lately introduced upon the American lines the truck takes its load, not upon double-inclined bearing surfaces, as in Fig. 115, but upon Davenport and Bridges' "swing-beam," already noticed.

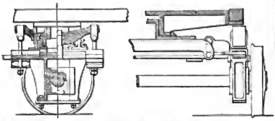

Fig. 116.—Bissell Truck, with Davenport and Bridges' "Swing-beam."

The author is unable to state whether the truck thus modified rides steadily. Mr. Hudson, of the Rogers Locomotive Works (U. S.), has adopted an arrangement whereby the truck, while free to move laterally, swivels freely also upon its own centre; and he has also introduced an equalizing lever between the truck and the springs of the front driving axle of coupled engines, thereby obtaining a better distribution of weight. It should be borne in mind that the principle of the Bissell truck is contained in Engerth's plan of coupling the engine and tender, already described.

The under framing required for the truck interferes in many cases with the room otherwise available for the leading wheels. It is often inconvenient, with the truck, to introduce wheels larger than 2 feet 9 inches, or 3 feet, where, without it, a diameter 1 foot greater, might be employed. On this account, at least, Mr. Adams' radial axle boxes, shown in Plate XXIII., are to be preferred. These boxes have the same freedom of vertical movement as the ordinary axle boxes, while they adapt their position horizontally to curves of whatever radii, in the same general manner as the two-wheeled Bissell truck. Geometrically, indeed, the two arrangements are identical; and in the case of engines with but a single pair of driving wheels, and having either the Bissell truck or the radial boxes under both ends, all the axles will radiate exactly to any curve when the king-bolt of the truck, or, in the case of the Adams' boxes, when their centre of curvature is midway between the driving and leading and the driving and trailing axles respectively. With coupled driving wheels, or with trailing axle boxes, guided in the ordinary manner, an exactly radial adjustment cannot be effected; but a compromise may be made by moving the king-bolt in one case, or the centre of curvature in the other, some distance nearer the driving axle. In Mr. Hudson's arrangement, already noticed, where the truck is free both to move laterally under the engine and to swivel around its own centre, no particular length of the bar governing the lateral motion is necessary, and it would be possible (although not desirable) to dispense with this bar altogether.

As two wheels of equal diameter, and fixed to the same axle, cannot roll around a curve without one of them slipping, and as "coning," if carried far enough to insure a proper adjustment of the diameters to different curves, would lead to great unsteadiness of motion, it has often been attempted to make the revolution of each wheel independent of that of the others, either by making one or both wheels of a pair loose on the axle, or by providing a separate axle for each wheel. The history of these attempts need not be given here, as no description of "loose wheels," other than Arnoux's already noticed, has been continued in regular service. The subject will be briefly discussed, however, in a further chapter.

The resistance opposed by a rolling load is greatly lessened by the application of springs. On good turnpike roads, Morin found that at a speed of 9 miles an hour the resistance to traction was twice greater without springs than with them. And whatever may be the difference in degree, the advantage from the use of springs is the same in kind on a railway as on a turnpike. In locomotives, however, a considerable proportion of the weight rests upon the rails directly, or without the intervention of springs. The wheels, tyres, axles, axle boxes, crank pins, a part of the connecting rods, and the whole of the coupling rods (together with the counterweights), and the eccentrics and eccentric hoops, are wholly unrelieved by springs. In some cases this literally dead weight amounts to one-third of the whole working weight of the engine. Not only is the resistance to traction greater for what may be called the unsuspended weight, but the wear of the tyres and rails is probably as much for 1 ton of this weight as for 2 tons of that hung upon the springs. The advantage of introducing a degree of elasticity between the axle and the road appears to have been understood at an early date. James Butler, in 1772, patented a wheel with spring spokes; and in 1831, Sir George Cayley proposed to ease the action between tyres and rails by introducing hard oak into a dove-tailed groove in the former, in such manner that the end grain of the wood should form the wearing surface. In 1837, Mr. W. B. Adams patented a "spring wheel," intended to afford increased ease of motion, as well as to prolong the wear of the tyres. In 1857, George S. Griggs, of the Boston and Providence

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* Levi Bissell's English Patent, December 1, 1858, No. 2,751.

Railroad (U. S), seated his engine tyres upon wood, in the following manner:—The rim of the cast-iron driving wheel was formed with a series of dove-tailed grooves across its width. As measured upon the circumference of the wheel, these grooves were 4 inches or 5 inches wide, and they were 1¼ inches deep, the bar or ridge of iron left between them being about 1⅜ inches thick. Oak blocks 1½ inches thick were driven into these grooves, and the wood surface (which rises beyond the iron ribs) was turned off in a lathe. The tyre was then shrunk directly upon the wood, and the whole quenched in water the moment the tyre was on, to prevent charring the oak seating. No tyre bolts were employed. This mode of seating engine tyres has come into extensive use in America, where it has been found that it increases their service one-half, their wear proceeding hardly more than two-thirds as fast as when seated on iron. On the Boston and Providence Railroad the Lowmoor and Bowling tyres of coupled driving wheels, 5 feet in diameter, and loaded to 3½ tons each, were formerly worn from 2¼ inches down to 1¼ inches thickness, after from 80,000 to 100,000 miles run. With wood bearings, the mileage of the same make of tyres, under the same engines, has increased to 140,000 miles. Mr. W. B. Adams, in 1858, introduced a mode of seating the tyre upon a hoop spring, confined in a groove formed in the rim of the wheel. This is shown in Plate XXIII. From the experience gained on the North London, Great Eastern, and the St. Helen's lines, it appears that Staffordshire tyres so seated last twice as long as the best Yorkshire tyres fixed in the ordinary manner; and considerably longer, also, than Krupp's steel tyres, when set without provision for elasticity. The principle of interposing an elastic cushion, like that in a horse's hoof, under the tyre, is indisputably sound; and it is only for experience to determine whether the hoop springs applied by Mr. Adams can be depended upon in continued service and at high speeds.

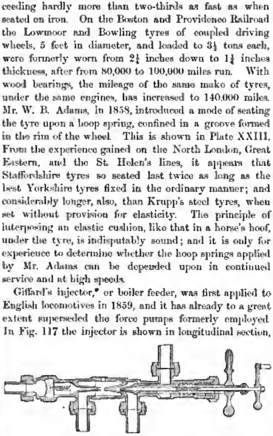
Fig. 117.—Giffard's Injector, 1858.

Giffard's injector,[*] or boiler feeder, was first applied to English locomotives in 1859, and it has already to a great extent superseded the force pumps formerly employed. In Fig. 117 the injector is shown in longitudinal section, and in a horizontal position. The pipe rising on the upper side admits the steam, the right-hand pipe on the lower side admits the water, the other pipe on the lower side is for the overflow of waste water, and the discharge is at the left-hand end of the instrument. Steam being admitted to the inner tube of the injector, and the central conical starting valve being withdrawn, the steam escapes in a jet near the top of the inlet water pipe. Supposing the level of the water supply to be below the injector, the escaping jet of steam, by its superficial action (or friction, as it may be termed) upon the air around it, forms a partial vacuum in the inlet pipe, and the water then rises in virtue of the external pressure of the atmosphere. Once risen to the jet, the water is seized, so to speak, and acted upon by the steam, in the same manner as the air had been seized and acted upon in first forming the partial vacuum into which the water rose. The action is that which Venturi, in the beginning of the present century, designated as the "lateral action of fluids," and after having been investigated to some extent by Dr. Young (vide *Natural Philosophy*, vol. ii., p. 534, 1805), it was proposed by Nicholson, in 1806 (*vide* p. 15 of the present work), for forcing water. It is the identical action of the steam jet in the locomotive chimney. It was employed in a great variety of apparatus schemed nearly half a century ago by the Marquis Manoury-Dectot;[*] it was employed more than twenty years ago to maintain the vacuum (or partial vacuum) in sugar-boiling pans;[†] and it was again employed some years since, for forcing water, by Du Tremblay and by Eugène Bourdon, in France; and in 1854 by Mr. Danchell, of Red Lion Square, London. Giffard's discovery was that the motion imparted by a jet of steam to a surrounding column of water was sufficient to force it into the boiler from which the steam was taken, and, indeed, into a boiler working at an even higher pressure. Thus the water in the boiler would rush out of an opening at the bottom with the same velocity as that with which water would escape from the bottom of a reservoir in which the head was such as to cause the same pressure per square inch. A column of water 2⅓ feet high passes upon its base with a force of 1 lb. per square inch; and if we take a boiler working at 100 lbs., the corresponding head of water pressure is 230 feet. The theoretical velocity with which water would escape, in the absence of every source of resistance, from under a head of 230 feet, is $\sqrt{230} \times 8{\cdot}01 = 121\frac{1}{2}$ feet nearly per second; and hence, in order that a jet of water may enter a boiler by its own momentum, against a pressure of 100 lbs per square inch, it must have a velocity greater than 121½ feet per second. In the present historical summary, an analysis of the entire action of the injector would be misplaced, and it is sufficient to observe that the fact of the great velocity of the water jet in the injector admits of clear and simple explanation, which will be given, in its proper place, in a further chapter.

It is an advantage of the injector that it may be worked while the engine is standing; and such, indeed, was the importance long since attached to a complete control of the supply of water to locomotive boilers, that independent feed pumps, or "donkey pumps," have for many years been employed in addition to the ordinary force pumps. The injector will not work when the feed-water is much hotter than 100°; and in the engines of the London and South-Western Railway, where feed-water heating apparatus is in general use, steam pumps are applied for service when standing, the ordinary pumps being retained for feeding the boiler when the engine is running.

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[*] H. J. Giffard's English patent, July 23, 1858, No. 1,665.

[*] Vide *Brevets d'Invention*, vol. xxvi., p. 88, Plates VII. and VIII.

[†] *Vide* Ewbanks' *Hydraulics*, p. 489.

In 1856, Messrs. Neilson,* of Glasgow, produced a single-cylinder locomotive of simple design for mineral traffic. It had four coupled wheels, the cylinder was horizontal, and placed under the fire box end of the boiler, the piston rod working through the after-cover, and a long cross-head, spanning the width of the engine, gave motion to a pair of connecting rods, which worked upon pins in the driving wheels. The coupling rods were made to act at right angles to each other, both to assist the engine over the centres and to prevent any chance of the connecting rods taking opposite directions at the commencement of a stroke.

In 1860, Mr. Ramsbottom, of the London and North-Western Railway, perfected and adopted means for filling the tenders of his engines with water while the train was in rapid motion. It had become necessary to run express trains long distances without stopping; as from London to Rugby, 82¾ miles, and from Chester to Holyhead, 84¾ miles. The supply of water necessary for such a journey varies, according to the weather, from 1,800 gallons to 2,400 gallons, the engines evaporating from 150 to 180 cubic feet per hour; and as it would be always necessary to be provided with the larger quantity, or nearly 11 tons of water, it became an important object to pick up a part of this on the way. Mr. Ramsbottom, therefore, had a pair of water troughs put down in a level part of the line near Conway. These troughs are of cast iron, 441 yards long, with a further length, at each end, of 16 yards, rising 1 in 100. They are 18 inches wide and 7 inches deep, and are placed one between the rails of each line, and so that the water, 5 inches deep, stands at about the level of the rails. The tender, of the ordinary pattern, and made to hold about 1,500 gallons, is provided with a rising water pipe, the lower end of which is fitted with an adjustable scoop 10 inches wide at the mouth. As this scoop meets the water, the effect is the same as if, the scoop being supposed at rest, the water were in motion at the velocity of the train. In either case the water must rise in the pipe, and, if permitted to do so, to a height the same as that from which a body falls, *in vacuo*, in acquiring the velocity with which the scoop and the water meet each other. At 32 feet per second, or rather less than 22 miles an hour, this height would be 16 feet. The height from the water trough to the top of the inlet pipe is 7½ feet, and a column of water of this height is just balanced, without delivery, at a speed of 15 miles an hour. The scoop dips 2 inches under the surface of the water, and at 22 miles an hour from 1,000 to 1,100 gallons are raised into the tender in 25 seconds. The delivery is practically the same at higher speeds, for although the velocity of the entering water is then greater, the time during which the apparatus is at work is diminished in the same proportion.

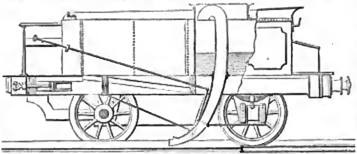

Fig. 118.—Ramsbottom's Self-Filling Tender, 1860.

A slight digression from the history of railway engines may be pardoned in the case of a locomotive made to draw trains of sledges upon ice. Fig. 119 represents an engine designed by Mr. Nathaniel Grew, and made by Messrs. Neilson and Co., and which was sent to Russia for the purpose of working on ice. Its weight was about 12 tons, the cylinders being 10 inches in diameter, stroke 22 inches, and diameter of driving wheels 5 feet. The tyres were studded with short steel spikes, to obtain sufficient adhesion. The engine was steered by hand gear acting upon the front end of the sledge. Mr. Grew had made a smaller engine for the same purpose in 1860; and this, as well as that shown in Fig. 119, was regularly worked on the Russian rivers, in the winter of 1861-62 in the transportation of passengers and goods.

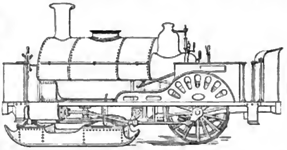

Fig. 119.—Grew's Ice Locomotive, 1861.

A number of locomotives of Lilliputian dimensions have been constructed within the last few years; and the dimensions and performances of these possess a degree of interest. Probably the smallest locomotive engine yet worked commercially is that (the "Tiny") designed and constructed in May, 1862, by Mr. Ramsbottom, and employed in the Crewe works of the London and North-Western Railway. A railway upwards of a quarter of a mile in length, and of only 18 inches gauge, connects the various parts of the works. It has curves of only 15 feet radius, corresponding to a radius of 47 feet for the 4-feet 8½-inch gauge. The "Tiny" has inside cylinders 4¾ inches in diameter, 6-inch stroke, and the four coupled wheels are 15 inches in diameter, and stand on a wheel-base of 3 feet. The boiler, 2 feet in diameter and 4 feet 6 inches long, has a fire box formed as a circular flue, 17 inches in diameter and 2 feet 5 inches long, while the tubes, 37 in number, are 1¾ inches in diameter and 2 feet long. The tube surface is 34 square feet, and the fire box surface 7½ square feet. A saddle tank on the engine carries 28 gallons of water, which is fed to the boiler by a No. 2 Injector. The whole length of the engine is 7 feet 7 inches, breadth 2 feet 6 inches, and the weight, in working order, is 2½ tons. Loads of from 12 tons to 15 tons are easily taken through the reversed curves of 15 feet radius.

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* *Vide* Walter M. Neilson's patent, April 24, 1856, No. 988.

In 1861, Mr. Isaac W. Boulton, of Ashton-under-Lyne, made a locomotive for a line of 2 feet gauge near Wigan. The railway is 1 mile long, and for 225 yards rises at the rate of 1 in 37, the remainder rising from 1 in 50 to 1 in 70, with several curves of 27 feet radius. For a considerable distance the line is in a tunnel only 6 feet 6 inches high. The engine has 5½-inch cylinders, 12-inch stroke, and is geared, in the ratio of 2 to 1, to coupled wheels 2 feet in diameter, and standing on a base of 3 feet 6 inches. The engine weighs 3 tons 12 cwt. The working pressure is 60 lbs. Its greatest load has been 22 empty wagons, weighing 12 cwt. each, up the steepest part of the line. The gravity of the whole weight of engine and train, 16 tons 16 cwt., upon an incline of 1 in 37, is 1,017 lbs., and adding 15 lbs. per ton for friction, the total resistance would be 1,269 lbs., which would be well within the utmost power of the engine.

An engine, of which several examples have been made by John Smith, of Coven, near Wolverhampton, for the Willenhall Furnaces, runs on a line of 2 feet 6 inches gauge, 1½ miles long, and having inclines varying from 1 in 30 to 1 in 70. The engine has 6½-inch cylinders, 12-inch stroke, and four coupled wheels 2 feet 6 inches in diameter, and standing on a base of 4 feet. The boiler is 2 feet 4 inches in diameter, the grate being 1 foot 10 inches by 2 feet 4 inches. The working pressure is 100 lbs., and the whole weight of the engine is rather less than 8 tons. It has taken a load of 17½ tons up an incline of 1 in 30, and regularly draws loads of 30 tons up inclines of 1 in 50 to 1 in 70 at the rate of 5 miles an hour, although, were the line longer, a higher rate of speed could be maintained.

The Neath Abbey engine, in the International Exhibition of 1862, was for a 2-foot 8-inch gauge; it had 8-inch cylinders, 16-inch stroke, and weighed 6 tons 17 cwt. when working in its ordinary trim.

An engine, the "Duplex," made by Mr. John Haswell, of the Austrian State Railway Works at Vienna, and exhibited in the International Exhibition of 1862, deserves a place in history. This engine, shown in Fig. 120, was designed to run with the utmost steadiness at high speeds; and for this purpose two cylinders, with their working parts, were provided for each driving wheel, and by connecting the pistons to crank pins diametrically opposite to each other, the whole momentum of each set of reciprocating parts was balanced by that of the other set. The pair of opposite crank pins is shown in Fig. 121. The distribution of the steam, at the same moment, to opposite ends of each pair of cylinders was effected by one slide valve, worked by the link motion. The engine had outside framing, and the journals of the driving axles were formed upon the bosses of the cranks. The four cylinders of the "Duplex" were 10¾ inches in diameter, equal to a single pair of $15\frac{3}{16}$ inches. The stroke was 24¾ inches, the driving wheels were 6 feet 9 inches in diameter, and the wheel-base was 11 feet 5 inches. The area of fire grate, for coke, was 15 square feet, the fire box surface was 84 square feet, and 160 tubes, $2\frac{1}{16}$ inches in diameter and 14 feet 6 inches long, presented 1,260 square feet of external surface. The working weight of the engine was 31 tons 14 cwt., of which 12 tons 6 cwt. were on the driving wheels. It was asserted that this engine ran with absolute steadiness at all speeds; but it is improbable that it was entirely free from rocking motion, although, as the constant upward thrust of the cross-heads upon the guide-bars (in running forward) was expended near the centre of gravity of the engine, the rocking was no doubt only slight. From all the experience that has been gained in counterweighing, it appears that the fore-and-aft motion of engines, as caused by the momentum of the reciprocating parts, may be as completely cured by revolving balance weights in the driving wheels as by the counter momentum of additional reciprocating parts of equal weight and moving in an opposite direction. In 1841 the late Mr. J. G. Bodmer patented a complicated arrangement, wherein two locomotive pistons were to move simultaneously in opposite directions in the same cylinder, the reciprocating parts attached to one piston counteracting the momentum of those attached to the other. All this was over-ingenious, yet the same reasoning appears to have been followed in the design for the heavy and complicated example shown in Fig. 120. In

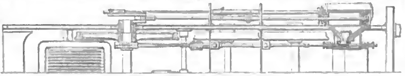

Fig. 121.—Haswell's Engine, Half Plan.

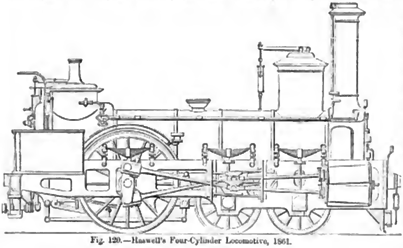

Fig. 120.—Haswell's Four-Cylinder Locomotive, 1861.

1847 one A. S. Livingstone* patented an engine, in which two cylinders were employed on each side, the connecting rods working upon crank pins, not diametrically opposite to each other, as in Fig. 120, but at points 90° apart in the circle of revolution. Here the reasoning was better than in Bodmer's and Haswell's plans, for it was Livingstone's object to prevent the rocking motion of outside-cylinder engines; and this would be effected when, as upon his plan would be the case, a constant and equal upward thrust of the cross-heads upon the guide-bars was maintained upon both sides of the engine. Livingstone was anything but a clever designer, however, for his cylinders were so placed that the piston rods were fixed, not in the *centres* of their respective pistons, but at points near their circumference!

The history of the Locomotive Engine has now been brought down nearly to the present time, and some of the preceding pages are, indeed, occupied with descriptions and criticisms of present practice. The author is fully aware how great is the difference between his own and other versions of the birth and growth of the Locomotive, more particularly with respect to the use of smooth driving wheels, the blast pipe, the multitubular boiler, the lap valve, the link motion, and the swivelling truck. Such evidence, however, as has been adduced in these pages will for the most part be found indisputable; and upon its examination the credit due for these inventions will be re-awarded by the reader of those popular industrial biographies which have lately created so many engineering heroes. The abundant reference which has been made to the Patent records has not been prompted by any over-estimate of the importance of a Patent *per se*, but because its specification affords, in most cases, an exact description, and always a precise date, which cannot be overlooked in histories of this kind any more than in patent litigation itself. There is merit both in the conception and in the original practical application of a useful invention; and in many cases the idea in which the whole novelty is contained is so crude that the difficulties in the way of reducing it to practice are such as to leave but a small share of credit to the inventor himself. But in the case of the various inventions under notice, it is to be borne in mind that most of them were successfully applied exactly in the form in which they were first designed.

A few conclusions which remain to be drawn from the history of the Locomotive Engine may now be stated, and for all practical purposes it will be sufficient if the historical period to which they apply be that between 1830 and the present time. To those who care for these conclusions it will be an object to know what locomotive history has taught, and, more especially, wherein it indicates the direction of future improvement. It has at least taught that the size of the Locomotive Engine may be increased with advantage to an extent not yet determined, and that it is susceptible of an almost infinite variety of arrangement. Seven-ton engines were reckoned heavy in 1830, and so were fifteen-ton engines in 1840, twenty-five-ton engines in 1850, and thirty-five-ton engines in 1860; yet there are goods engines now at work on the Continent which weigh, exclusive of tender, 45 tons; and while English engineers appear disposed to approach this limit, M. Meyer's design of 1862, which has been much commended, contemplates a weight of 60 tons, exclusive of tender. These great weights have been provided for in part by increasing the load upon each wheel, and for the rest by adding to the number of wheels. Three tons to each driving wheel was the limit in the engines of 1830. By 1850, 6 tons upon a single wheel were allowed, and although this load is not often exceeded, 7½ tons on each driving wheel is permitted in the Caledonian express engines, and 9¼ tons was tolerated in the first engines made for working the Bhore Ghât incline of the Great Indian Peninsula Railway; while some of Engerth's engines, on the South Austrian line, at one time had nearly 10 tons on each wheel, in the case of the hind wheels of the tender. These great weights, it is to be remembered, have been permitted with wrought-iron tyres and iron rails, but with Krupp's or other steel tyres and Bessemer rails it would be premature to say that even 10 tons upon a single wheel, or 20 tons on an axle, may not fall within the limit of working economy. Not only is there much to hope from steel tyres and steel rails, but still more from elastic seating to the tyres, either upon Mr. W. B. Adams' or Mr. Griggs' plan, and as much more from a good system of slightly elastic permanent way. There are various objections to a large number of wheels under an engine, especially when they are coupled; and, of course, the greater the weight which, with a fair regard for economy, can be carried upon each wheel, the less the number of wheels required for an engine of a given total weight. As for the number of wheels which can be practically worked under an engine, we find examples of four, six, eight, ten, and twelve; and, indeed, in many cases, these numbers of wheels have been coupled in the same engine.

As for variety of arrangement, examples have been given of engines with one, two, three, four, and five cylinders, both outside and inside, and placed vertically, horizontally, and in an inclined position. There are horizontal and upright boilers, and boilers with fire tubes and with water tubes, and with an almost infinite variety of structural features. The details are even more various, and it has been argued by certain locomotive engineers that any design, whether of detail or of an entire locomotive, may be made to answer in practice, if it be well executed. Not, however, that, amid this variety, one plan is not better than another. Practically, the locomotive boiler, as first used, is still found to be the best; and it is nearly as applicable with coal and with wood as with coke, care in firing being found to serve the purpose of most of the mechanical contrivances designed to assist the combustion of the former. The link motion is found to be, practically, the best form of valve gear, and thus the more important members of the locomotive engine still remain essentially the same as they were twenty years or more ago. The knowledge has been gained, however, that the evaporative economy of the boilers of railway locomotives is equal to that of the best form of land boilers; and also, that locomotive boilers can economically evaporate very variable quantities of water in a given time, according to the frequency of the escape of exhaust steam up the chimney. Thus while, in the case of two boilers of identical dimensions, one of them may not, in a goods engine, evaporate more than 100 cubic feet of water per hour, the other, in an express engine, may economically boil away twice this quantity.

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* A. S. Livingstone's Patent, No. 11,845, August 23, 1847.

It has been found that a maximum adhesion, upon a clean dry rail, of three-tenths, and even three-eighths, of the weight on the driving wheels is occasionally attained. Although this is much more than has been counted upon by engineers, it is even less than might have been inferred from Morin's and Rennie's experiments, which show a co-efficient of friction, between wrought iron and wrought iron, of about 45 per cent. of the force with which they are pressed together. A better knowledge of the extent of the adhesion between the driving wheels and the rails has led to the working of steeper and steeper inclines, until, as an extreme effort, loads have been taken, in practice, up gradients of 1 in 10; and no inclination less steep than 1 in 40 is now considered a serious obstacle to the practical working of a large traffic.

It has been gradually learned that a very low centre of gravity is of but little importance in a locomotive engine; and that when the cylinders are horizontal, well fastened, and well covered, they may as well be placed outside as within the main framing; and that proper counterweights will correct the fore-and-aft and the sinuous motions which the earlier engines so commonly manifested.

It is not yet settled what is the greatest steam pressure that can be carried economically. Apart from considerations of safety, it does not quite appear that steam of a pressure greater than 130 lbs. per square inch can be worked with advantage, when the proper lubrication of the cylinders and valve faces is considered. But up to this limit a decided advantage has been heretofore found in increasing the pressure of steam from the 50lbs. per square inch formerly employed. On the other hand, the maximum economical speed of piston has not been ascertained; and it would appear that, with a high piston speed, small driving wheels and light engines are preferable to the colossal examples with which recent practice has made us familiar. Small-wheeled engines have been found to start a train more rapidly, and to draw it with greater regularity of motion, than engines with from 6½-feet to 8-feet driving wheels; and a comparatively small boiler, with the greater frequency which a small driving wheel gives to the blasts of the waste steam, will evaporate as large a quantity of water, in a given time, as a much larger boiler when working in connection with a larger driving wheel. Except in connection with the frequency and force of the blast, there is no definite limit to the evaporative efficiency of a locomotive boiler having a given amount of heating surface; for it has been proved that large boilers working with large wheels do not evaporate a greater quantity of water per hour than smaller boilers working correspondingly smaller wheels. With easy springs, and with tyres which, because of their hardness, preserve their cylindrical form, a 5-feet wheel may be economically worked at 60 miles an hour as well as a wheel of nearly twice the size. And as for the speed of piston, it is not yet determined what is the limit beyond which it may not be economically worked—a continuous mean speed of 1,200 feet per minute at least being practicable with advantage.

With the Bissell truck or with radial axle boxes, much greater flexibility of action in curves is secured, and a longer wheel-base may be also ventured upon. The advantage of a long wheel-base is that, so long as it will ride through the curves, it affords a better distribution of weight, and secures greater steadiness of motion than when the total span of the wheels is comparatively short.

Increasing experience gives support to the probability that the tender will yet be wholly dispensed with. Few engines burn more than half a ton of coal or coke per hour, and thus 1 ton generally suffices for the longest continuous run. The practicability of picking up water from open troughs laid down on the line having been already proved, not more than 2 or 3 tons of water, sufficient for half an hour's run, need be carried in most cases; and any engine can easily bear the weight of 3 or 4 tons of supplies, where a ponderous vehicle ordinarily weighing, when fully loaded, from 15 to 23 tons, is thereby dispensed with. The tender, even if it carried no load, must have great strength and consequent weight, in order to hold together, inasmuch as it may be regarded as a distance-piece between the engine and train, and as subjected to the whole force of the former, and often to severe shocks from the latter, to say nothing of the trial imposed by running at great speed over a line which is only approximately smooth where it is not positively rough. Besides the weight of the tender, which often absorbs from 60 to 80 indicated horse-power in drawing it at high speed, this very effort involves from 2 to 3 tons of additional weight in the engine, above what would be sufficient if the tender could be dispensed with, as, in an improved practice, it is likely to be.

CHAPTER IV.

HEAT AND STEAM.

The Latin word, *ingenium*, besides signifying heart, mind, abilities, or genius, was originally applied to any mechanical device or contrivance. In this sense *ingenium* became Anglicised into engine, and those who made or worked mechanical contrivances came to be called engineers. The word, engine, has been still further contracted into "gin," as in the case of the cotton gin, which is, strictly, an engine for cleaning cotton. It is more usual, however, to restrict the application of the term engine to machines in which motion is produced by a natural agent; and even here, among prime movers, it is unusual to speak of "water engines" or "wind engines," the term being more especially reserved for those worked by steam, air, or gas, and (as has been done experimentally) by electro-magnetism. Locomotive Engines, so called from the Latin, *locus*, a place, and *motio*, motion, are engines having the power of moving themselves from place to place. For the purposes of the present treatise, they are to be considered as Steam Engines having the power of self-movement along railways.

As a steam engine, the Locomotive should be studied with the aid of a knowledge of the nature and properties of steam. Steam (in the earlier writings, *stem*) is a Saxon word, originally denoting only the visible emanation from warm or boiling water. Visible or adynamic "steam," however, is but an assemblage of suspended watery particles, of atmospheric temperature, and less than atmospheric energy. True steam is invisible; and although the original signification of the word "steam" is still preserved in popular and colloquial phrase, its only application in the language of science is to that expanded or super-liquid state of water in which, irrespective of its own gravity, it is capable of exerting mechanical power. The Latin word *vapor* had originally the same meaning as steam, but, Anglicised into "vapour," it is now commonly employed to denote such aëriform bodies, of whatever nature, as are condensable without extraordinary pressure or extreme cold. Thus we have the vapours of alcohol, bromine, mercury, &c., as well as steam, the vapour of water. The terms steam and vapour are interchangeable only when applied to the aëriform state of water, usage forbidding such an expression as "the steam of camphor," although it doubtless admits of logical justification. It would be superfluous, therefore, to speak of "the steam of water," for no other steam is understood. Steam already condensed in a visible voluminous mass in the atmosphere is often, but inaccurately, described as "aqueous vapour." In reality it is only gravitating and confluent water, while, furthermore, true steam is in its nature aqueous, just as the vapour of sulphur is sulphurous, "Mist," "fog," "cloud," or "spray," applied to so-called "aqueous vapour," would answer every purpose of ordinary description. Steam is sometimes described as a gas,* although chemistry reserves this term for those aëriform bodies which are only condensable under great pressures or extreme cold. So long, however, as steam retains the full quantity of heat indispensable to its continuance in the aëriform state, it is amenable, with some qualifications, to the laws of gaseous bodies.

In common phrase, water, when solidified by privation of heat, becomes ice, and when vaporized by access of heat, becomes steam; and in either case it no longer remains as water. Chemistry, however, recognizes only an identical combination of hydrogen and oxygen (HO) in these several conditions, which may be defined, therefore, as the solidiform, the aquiform, and the aëriform states of water. We have other liquids, as mercury, sulphuric acid, and various oils, which may be both solidified and aërified; certain solids, as gold, which may be not only melted, but converted into vapour; and a few so-called permanent gases, which may be liquified, while carbonic acid gas may be even solidified. In these changes of condition, heat is recognized as the active agent in producing liquefaction and aërifaction, while a greater or less abstraction of heat by compression or radiation causes solidification. Water, air, sulphuric ether, liquid carbonic acid, and many other substances, are highly expansible under the influence of heat; but in all cases where they are thus expanded for dynamical purposes, the energy of expansion, and, consequently, the mechanical work obtained, are due to heat alone, steam and all other elastic vapours being simply the vehicles of heat. Watt understood this fact so well that he described his great invention as a "fire engine;" and the comprehensive term "heat engine" is now applied, by the first authorities in science, to all machines from which mechanical power is obtained by the progressive expansion, or alternate expansion and contraction, of any substance whatever. Any inquiry, therefore, into the nature and properties of steam ought to be first directed to the general mode of action and mechanical laws of heat.

The original principle of heat is, of course, ascribable only to Creative Power, in the same manner as gravity, light, electricity, and life itself. Neither of these phenomena is explicable by facts, nor upon merely material hypotheses, and any philosophical interrogation with this object must fail. Philosophers in all ages have, however, endeavoured to detect the earliest physical circumstances which attend the manifestation of heat, and thus to eliminate, as supposed causes of heat, merely intercedent or secondary effects. The doctrine of combustion, commonly attributed to Lavoisier, but virtually propounded by Dr. Hooke in 1665, and again in 1676; Dr. Black's subsequent discovery of "latent" heat, and the final adoption of the atomic theory, have given great precision to our knowledge of certain properties of heat; but its cause yet remains unascertained. Two hypotheses under this head—neither of them, probably, susceptible of proof or disproof—have, however, divided philosophers for centuries. One is, that heat is in itself a material substance, although one of exceeding tenuity—permeating and enveloping the molecules or atoms of all other substances, while the imaginary particles of heat are furthermore supposed to be highly repellant of each other. This supposed material substance of heat is designated as caloric (from the Latin *calor*, the same as *heat* in our own tongue), to distinguish it from the physical sensation so commonly mistaken for heat itself. This hypothesis of the materiality of heat has been very generally entertained. The other hypothesis is, that heat consists in exceedingly minute and rapid motions of the ultimate atoms or molecules of all kinds of matter. This opinion was held by Galileo, Bacon, Boyle, and Newton, and at a later period by Rumford and Davy, and is maintained by many eminent philosophers and physicists at the present day. It is not easy to grasp, mentally, the idea of a molecular motion corresponding to the known effects of heat. Black, in his lectures, observed upon this point—"I cannot form to myself a conception of this internal tremor." In the article "Heat," in the last edition of the *Encyclopædia Britannica*, it is observed that "This opinion, however (that of molecular motion), seems vague and unsatisfactory. If we say that heat is motion amongst the particles of matter, still we have no explanation of the manner in which this motion is produced, for we cannot conceive any movement without an impulse, nor an impulse without a material agent. Heat pervades all sorts of matter; it remains in some circumstances dormant, or, as it is termed, latent, and may be again elicited from bodies by various means. Did it consist in vibrations or motions of the particles of other matter, it should pervade elastic bodies with the greatest celerity, which we know not to be the fact. It will, for instance, pervade a rod of lead, or of the softest copper, far more readily than an equal length of glass or of marble." Black observed, also, that, in apparent opposition to the motory hypothesis, the hammer-blows which quickly produce heating in soft iron, hardly affect the temperature of steel, which, from its superior elasticity, might be supposed to admit of a rapid propagation of motion among its particles.

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* Gas, from the Saxon *gast*, corresponding to the German *geist*, and equivalent to the English word *spirit*, or supernatural power, form, or action. The word *ghost* is thus derived.

Another and a recent hypothesis, constructed apparently from the two already described, is that of nuclei of ether surrounded by "elastic atmospheres" of matter, which "atmospheres" are supposed to be either revolving or vibrating about the ethereal nuclei which they envelop. This is the "hypothesis of molecular vortices," and it has been framed in an ingeniously bewildering manner, so as to admit of a motion of the nuclei themselves, this motion, intended to account for radient heat, being furthermore supposed to be sometimes imparted to, and at other times received from, the constantly whirling or vibrating "atmospheres."

All these hypotheses are founded upon the familiar and almost inevitable supposition of the existence of substantial matter—oxygen, nitrogen, gold, iron, &c. (or, more precisely, the sixty-five elements now known to chemistry), being regarded as matter. Newton believed—and most philosophers since his time have adopted his opinion—that matter consists of atoms, each exceedingly small, and in itself absolutely impenetrable and unchangeable. It is true that no subdivision of any substance whatever has yet resulted demonstrably in the extrication of an atom absolutely incapable of division by itself. Thus, a single grain in weight of metal, in a dilute solution, has been divided, under the microscope, into seventy thousand millions of parts, without apparently reaching the limit of ultimate divisibility. It is a fact, however, that, in all chemical combinations,—as in the union of hydrogen and oxygen in the formation of water—carbon and oxygen in the formation of carbonic oxide and carbonic acid—nitrogen and hydrogen in the production of ammonia, &c.,—different substances unite only in definite proportions, irrespective of the quantity of each or both which may be present; and this fact almost irresistibly leads the mind to the conclusion that there *is* a limit to the divisibility of matter.

Boscovich, however, in the last century, taught that there was no such thing as substantial matter, and that apparent substance—iron, for example—was but an aggregation or collection of "centres of force." It is not easy to admit this hypothesis into the mind, yet Faraday has inclined to it; and in a metaphysical consideration, proceeding in the manner of Bishop Berkeley, as much may perhaps be said in favour of as against non-materiality. Boscovich's notion has enabled Faraday to reconcile some otherwise unexplained electrical phenomena; but in thus arbitrarily dismissing the sixty-five supposed elementary divisions of palpable and ponderable matter, and at the same time admitting sixty-five distinct forms of force in addition to those which we have been accustomed to consider, it is questionable whether we shall have greatly advanced in our inquiry into the sources and effects of physical action.

Whichever hypothesis of matter or of heat be entertained, the imaginary phenomena which it adopts are, of course, beyond, not only the direct perceptions of our senses, but also any other means of identification yet known. It is apparently impossible, therefore, to reduce either hypothesis to fact, and thus the mind may incline to the rejection of these, if not all speculations, as to the true character of heat. But in the absence of actual knowledge, even an assumption, presenting that degree of probability, or accordance with observed facts, which can commend it to an inquiring and discriminating mind, is very desirable. If we can be satisfied either with the assumption of substantial caloric, or that of molecular motion, many of the most striking phenomena attending the generation of steam and its action in steam engines may be readily explained. Without some fundamental notion of matter and heat, we can form no rational conception of so-called "latent" heat, specific heat, nor, indeed, of combustion, fusing, welding, liquefaction, vaporization, condensation, solidification, and other ascertained effects resulting either from the reception or emission of heat.

To prepare the way for an intelligible conception of heat, we must first clearly distinguish between its cause and the bodily sensation which it produces, and then dismiss all further thought of the latter from our minds. Physically, then, our own bodies are affected by heat in the same manner as inert substances. The material particles which compose our frames are forced by it to increased distances from each other; the water especially, of which all animal bodies are largely constituted, is collectively dilated and in part evaporated; while, if a sufficient degree of heat were applied, decomposition would result. When these effects are produced upon a lifeless body we cannot contemplate them as being attended with any sensation whatever. In the living frame, however, and which, as a material thing, chemically considered, must be identical with the same body after life has just departed, heat produces sensations which are mild or acute according to the temperature applied. This is simply the result of vital sensibility, and not necessarily of anything in the character of heat, whereby it differs from mere force. We may conceive at least the acute sensation producible by great heat, if we imagine each particle collected under even a small surface of our bodies to be forcibly thrust apart from those surrounding it. The finest puncture made by a needle produces a sensation which may be reasonably regarded as identical with that which would result from heat communicated to the same particles of the body, and to no others; and, indeed, there is no difference in the sensation caused by a slight puncture by a needle, whether it be at the ordinary temperature of the air, or whether it be previously heated to a white heat. Nitrate of silver does not increase the temperature of the parts of the human body to which it may be applied, and does not burn, therefore, in the manner of a hot iron; yet it produces the identical sensation of burning, by the forcible decomposition of the living tissues, which are torn apart, atom by atom, by chemical force, in obedience to the affinity which some of these atoms have for the caustic applied, an affinity greater than the mutual force of cohesion of the atoms of the flesh itself.

The readiest, and perhaps the most satisfactory, conception that can be formed of the mutual relations of matter and force is that which pictures the former as consisting of innumerable minute particles, each absolutely indivisible, and each of which forms the centre both of a class of attractive forces—corresponding to gravitation, the attraction of cohesion, and in some cases of chemical affinity and magnetism,—and of a class of repulsive forces represented by heat and electricity, the latter being the most powerful decomposing agent known to science. It is possible that the forces of gravitation, cohesion, chemical affinity, and magnetism are but different manifestations of one and the same force, and that heat and electricity are identical in their nature.* The action of heat will be best understood by regarding it as a repulsive force simply, acting equally in all directions from the atom in which it is centred, the atom being supposed at rest. As all matter possesses some heat, and thus some repulsive force, the atoms of matter can never, upon this view, be in actual contact with each other. In themselves we must consider them unchangeable in form or size by any agency whatever. Their distance from each other, in the same body, will be determined by the relations of the opposite forces to which they are subject. If the force of cohesion greatly exceed that of heat, the atoms will form a solid body, but still, we must suppose, without being in actual contact with each other. A further accession of heat may so far loosen the attraction of the atoms for each other as to permit of their rolling freely among themselves, as in the liquid state of matter. Here the heat, or repulsive force, must be supposed to have removed the atoms conceivably further from each other than when they cohered as a solid. This supposition is not necessarily opposed to the fact that ice has a sensibly larger volume than the water from which it congealed; for the particles of ice, like those of snow, consist of crystals, in which, by the very force of crystalline arrangement, minute interstitial spaces are left between the atoms. Bismuth, too, expands in cooling, and solid iron will float upon the melted metal in the founder's ladle. A further accession of heat, completely overcoming the force of inter-atomic attraction, and thus removing the atoms indefinitely further from each other, leaves them in the gaseous state of matter. We have no authority for speaking of the atoms as liquid atoms and vapour atoms, and must consider them, therefore, as solid atoms, in all the conditions of matter alike, these conditions, as already stated, depending only on the mutual relations of the opposite forces of the atoms. In chemistry we can, and indeed must, conceive every known elementary substance as capable of taking the solidiform, the aquiform, and the aëriform conditions, without any change in the nature of the substance itself, or in the form, size, or weight of its ultimate atoms.

This consideration of matter and force involves the proposition that the repulsive force of heat, unlike all the known attractive forces, may in part permanently forsake one atom for another. All notions of heat require some stretch of the imagination; but we are nevertheless justified in indulging the imagination to some extent in this case, because, so long as we do not assume what is known to be impossible, we may thus arrive at better views of the action of heat. Now, it is as easy to imagine the transference of a force from one atom to another, as the transference of a fluid answering to the old notion of caloric, and as easy as to imagine the propagation of molecular motion, which, although based upon familiar analogies, nevertheless implies also a transference of force, partly as now assumed, while it fails to explain many of the phenomena of heat. Thus no explanation can be given, upon this hypothesis, of the heat generated by combustion. An atom of oxygen and an atom of carbon or of hydrogen may leap towards each other and unite by chemical affinity, but the heat liberated cannot be supposed to be due to increased molecular motion of the compound atoms *after* they have, so to speak, been locked together.

Considered as force, heat, like other forces, may be measured quantitatively. The thermometer indicates only the intensity of heat, just as a steam-pressure gauge indicates the intensity of the pressure of steam in a boiler, without, however, in any way representing the quantity, as volume or total weight, contained therein. But exactly as one boiler twice as capacious as another must contain

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[*] Vide, *An Inquiry into the Nature of Heat, and into its Mode of Action in the Phenomena of Combustion, Vaporization, &c.*, by Zerah Colburn. London, E. and F. N. Spon, 1863.

twice as much steam, of a given pressure, so 2 gallons, or 2 pounds of water, or of oil, at a given temperature, must contain twice the quantity of heat present in 1 gallon or 1 pound of the same substance at the same temperature. A convenient and now generally adopted unit, for the measurement of quantities of heat, is that, the heat, required to raise the temperature of 1 British pound of pure water 1 degree Fahrenheit in temperature. The French thermal unit, or *calorie*, is similarly the heat required to raise 1 kilogramme of water 1 degree Centigrade. These units are respectively adopted upon the assumption that, while the quantity of heat required to raise 100 lbs. of water 1 degree is necessarily one hundred times greater than would be required to warm 1 pound of water to the same extent, the heat required to raise the temperature of 1 pound of water 100 degrees is also one hundred times that required to warm it 1 degree. But this assumption is not quite correct; for, although the mean temperature resulting from the mixture of equal weights of water at say 50 degrees and 150 degrees respectively, is commonly supposed to be 100 degrees, it is in reality somewhat more; thus proving that a greater absolute quantity of heat is required to raise 1 pound of water from say 149 degrees to 150 degrees, than from 49 degrees to 50 degrees. Hence the thermal unit, or the heat required to raise 1 pound of water by 1 degree Fahrenheit, applies by common consent to water at its maximum density, corresponding to 39 degrees.

So far the thermal unit is an arbitrary measure; but we shall presently consider its relation to mechanical work; for, whether heat be regarded as in itself force, or as a manifestation of force in the form of molecular motion, it must, like other forces, be convertible into and measurable by mechanical work. Of the whole elastic force of steam—a force utilized in mechanical work in every steam engine—successive portions must have been imparted by successive increments of heat; and thus the heat first expended in raising the water to the boiling point must, as it is precisely the same in *kind* as the heat expended in evaporation, have a definite mechanical value, say in foot-pounds of work.

Not only do the indications of the thermometer, or what may be called thermometric heat, bear no relation to the quantity of matter to which the thermometer is applied, but they do not, either, express the quantity of heat which may be present in a given quantity of matter, as 1 lb. of water. The mere fact that a body is sensibly hot is no proof of the quantity of heat present. Thus, a pound of iron, at a bright red heat in daylight, contains not much more heat than an equal weight of water at about the highest temperature employed in bathing. The proof of this consists in the fact that 1 pound of iron at say 1,200 degrees will not, on being plunged into an equal weight of ice-cold water, raise its temperature beyond 149 degrees or 150 degrees, the temperature of the iron falling, on immersion, to that of the water. So, too, true steam may exist at a temperature of even 32 degrees, and it does exist, diffused in the atmosphere, at an even lower temperature. It has been the custom to speak of "moisture" as being "suspended" in the atmosphere; but all the watery particles forming cloud must have risen, as steam of low tension, from the earth; and in the clearest atmosphere a certain quantity of moisture may be collected, corresponding always to the density of true steam of that temperature, the steam rising to the exact height in the atmosphere where its own weight balances its elasticity. Yet the steam which rises from water a little more than "blood-warm," or at say 102 degrees, has nearly as much total heat, or is capable, in proportion to its own weight, of heating nearly as much water, say 1 degree, as steam of the highest pressure worked in locomotive engines.

The relative quantities of heat which different bodies contain at the same temperature may be tabulated as follows, from the experiments of M. Regnault:—

	Specific Heats.		Specific Heats.
Water, . . .	1·0000	Mercury, . .	0·0333
Iron, pure, . .	0·1138	Cast Iron, . .	0·1298
Copper, . .	0·0951	Steel, . .	0·1185
Lead, . . .	0·0314	Carbon, . .	0·2411

These numbers will be thus understood:—only 0·1138 of the quantity of heat which suffices to raise 1 lb. of water by 1 degree is required to warm 1 lb. of pure iron to the same extent. In other words, the heat that will raise the temperature of 1 lb. of water 1 degree will raise that of 1 lb. of iron nearly 9 degrees, or, conversely, 9 lbs. of pure iron 1 degree. The quantity of heat necessary to raise the temperature of 1 lb. of water 1 degree will raise that of 30 lbs. of mercury to the same extent. It is one of the phenomena of heat, of which we cannot know the cause, that in different substances it collects around, and is exerted from, the atoms in very different quantities at the same intensity or temperature. The fact of the passage of heat from one body to another depends, however, not upon the quantity of heat present, but upon its temperature or intensity. A pound of steam of a temperature of 212 degrees has more than four times the total quantity of heat contained in a pound of iron at 220 degrees, with which it may be in contact. Yet here the steam will receive heat from the iron, although the rise in the temperature of the steam will not proceed at the same rate as the fall of the temperature of the iron—the specific heat of steam being greater than that of iron, although it is much less than that of the water from which it was generated. These relations of the quantity and temperature of heat will be better understood from the consideration of the relative quantities and pressure of different volumes of steam. Thus with two boilers, one ten times more capacious than the other, the larger one, filled with steam of 20 lbs., will receive steam from the smaller one if it contain steam of 25 lbs. pressure. Or, to give another illustration, 10 lbs. of water at 50 degrees would receive heat from 1 lb. heated to 55 degrees. Yet we cannot say that the pound of water in the smaller boiler contains more heat or steam than that to which a delivery is actually made.

Wilcke originated the term specific heat, to express the relative total quantities of heat acquired by different substances with equal increments of temperature, as already explained; but it is to be understood that, in the passage from the solid to the liquid, or from the liquid to the gaseous state, a quantity of heat, which in the case of most substances is very large, is appropriated, but which appropriation is never described as "specific heat." Thus the specific heat of ice, according to M. Person, is 0·504, or half that of water; one half as much heat being required to raise 1 pound of ice, from say 30 degrees to 31 degrees, as enters 1 pound of water in raising it from 39 degrees to 40 degrees. But ice melts at 32 degrees, provided heat be supplied for the purpose; for although the temperature does not rise by liquefaction (the water remaining at 32 degrees), yet in order to melt 1 lb. of ice as much heat is required as to raise the temperature of 1 lb. of water 140 degrees. In other words, to melt 1 lb of ice by 1 lb. of water (the final temperature of both to be 32 degrees), the water must be first heated to $32° + 140° = 172°$; or if 2 lbs. of water be employed, it must be heated to $32 + \frac{140°}{2} = 102$ degrees. We do not here say that the specific heat of the ice has increased, although a large quantity of heat, of which the thermometer gives no indication, has indisputably been acquired. Dr. Black gave to the heat which would thus seem to have disappeared the name of "latent heat;" and although the term has been often and justly objected to, it is still employed. If we consider heat, as we now propose to do, as force simply, we find the "latent" force fully expended in overcoming the cohesion of the particles of the ice, and in holding them apart in the liquid state. We may conceive each atom as surrounded pictorially by a visible atmosphere of force, and that the flow of heat-force from one atom to another is perceptible only at the exteriors of these atmospheres. The heat necessary to any indication upon a thermometer would then proceed from the exterior only of these imaginary atmospheres of force; and thus the mere intensity of the heat apparently absorbed in the liquefaction of ice would correspond to but 32 degrees, although, to keep up the ideal illustration, the full intensity of the heat of the substance employed to liquefy the ice might be found within this atmosphere of force, if the particles of mercury in the bulb of the thermometer could only be placed there. It is to be remembered that something warmer than ice must be employed to melt it: for instance, water at exactly 32 degrees would never melt it, although with the slightest conceivable addition to this temperature liquefaction would be but a question of time and of the quantity of water available for the purpose.

Such views of the action of heat are not to be too lightly valued. Beyond the numerical equivalents determined for its various effects, we as yet know nothing of this almost omnipotent agent, except what an instructed imagination presents to us. No intelligent mind can remain satisfied without some hypothesis of heat, and the proofs against the material, or caloric, and the motory hypotheses are so numerous that even an idealization of transferable force is not to be cast aside as impracticable.

In evaporation, whether from the lowest stratum of a liquid mass, or from the surface by atmospheric or other heat, the hypothesis of heat-force explains what is hardly, if at all, explicable upon other views. So great, indeed, has been the difficulty in reconciling to these views the process of vaporization, that an ingenious essay has been lately written to show that there is no such thing as heated liquid water; or, in other words, that water can only exist in the three states of ice, a liquid at 32 degrees, and as vapour; and that water apparently warmer than 32 degrees is but a mixture of water of that temperature with steam, of which the temperature is not given. This view contains a self-evident fallacy, inasmuch as, if steam is only reconvertible into water of 32 degrees, no surface warmer than 32 degrees could possibly condense it—a conclusion which is diametrically opposed to fact. The appearance of such a speculation, however, and its dissemination to a considerable extent, show the importance of an hypothesis of heat which, however incapable of positive proof it may be, shall not at least do violence to familiar facts.

On the view of heat-force, the least conceivable intensity of this force beyond that corresponding to 32 degrees will liquefy ice, and any excess beyond the total heat of liquefaction will be subsequently expended in the bodily expansion of the liquid water. The particles of water have, however, a strong attraction for each other—so strong, indeed, that it may be proved that less heat is required to melt a pound of iron than to tear apart the molecules of an equal weight of water into steam. For, while 1 pound of coal will not vaporize more than 9 lbs. of water in practice, a pound of good founder's coke will, in the best cupolas, melt down 16 lbs. of pig iron. Although the apparent cohesion of water is nothing, and that of ice very great, far more power is requisite to completely separate the particles of the former than those of the latter. For the only mode in which the particles of water can be fully separated is by vaporization, and we have an exact measure of the force, or rather the mechanical work, required to effect this, inasmuch as we know the quantity of heat required. It is not certain at what extreme distance from each other the particles of water continue to exercise a sensible mutual attraction, but it is very great, as may be shown. Thus, the steam (for it is steam, notwithstanding its slight pressure) rising from water at 32 degrees occupies 187,407 times its original volume as a liquid, its particles, considered as centres of force, being removed, therefore, to $\sqrt[3]{187,407} = 57$ times their original distance apart. Yet, until this great comparative removal has been effected, the particles continue under the thrall of mutual attraction, and are not vaporized by a temperature of 32 degrees. As we rise from a temperature of 32 degrees, the steam produced is of greater density and pressure, and as the particles are thus nearer to each other than in the case just considered, their mutual attraction is still greater, and a further total quantity and greater intensity of heat is required to hold them apart. That most of the so-called latent heat of evaporation is expended in overcoming the force of attraction of the particles of the liquid is easily shown when we know the mechanical value of the heat required to warm 1 pound of water 1 degree. At the risk of anticipating the substance of another portion of the present chapter, it may be here said, that it is fully proved that 772 foot-pounds of mechanical work are represented by the British thermal unit, whether the work be obtained in cooling or expended in warming, to the extent which that unit represents—viz., 1 pound of pure water × 1 degree Fahrenheit. For the present, this fact may suffice to prove how much of the heat of evaporation goes in overcoming the attraction of the particles of the liquid. Thus, in evaporating 1 pound of water into steam of atmospheric pressure, 26·35 cubic feet are formed; and we may consider this as a column 1 foot square at the base, and rising, in the manner of a piston, 26·35 feet high, against the atmospheric pressure of 14·7 lbs. per square inch. Now, the erection of this imaginary column, against this pressure, constitutes the whole of the external work performed by the heat in evaporating 1 lb. of water into steam of 212 degrees. The work done is, therefore, 144 × 14·7 lbs. =2,116·8 lbs. through 26·35 feet, or 55,778 foot-pounds. But as the heating of 1 lb. of water 1 degree corresponds to 772 foot-pounds, this quantity of work is equivalent only to heating 1 lb. of water 72 degrees; whereas, if the water were originally at 32 degrees, we know that 1,146 units of heat must have been expended in evaporating it. Of the whole heat, therefore, 1,146°—72°=1,074°, or 1,074 thermal units, can only be accounted for by referring to the molecular attraction of the particles of the water for each other.

In this way the disappearance of so-called latent heat may be readily understood. In steam it is a force fully occupied by an opposing resistance, and hence powerless to impart to the mercury in a thermometer the force requisite to expand its particles visibly upon the scale beyond the point corresponding to the sensible temperature of the steam. But let a colder object be presented, and the heat-force flies to it at once, dismissing the particles previously existing as steam, and thus leaving them to unite, by their own attraction and by gravitation, into liquid water. There is no loss of heat, in a physical sense, in the return from steam to water; and it is for engineers to find how they may recover the whole of it in mechanical work, or, to use the common term, "commercially."

The process of vaporization may now be understood in a manner affording a clearer view of the relations of heat and water than could be found in the mere quantitative results to which most writers are in the habit of referring for all that they wish to convey on the subject of steam. A quantity of say ice-cold water contained in any vessel may receive heat from below, or at the sides, or at the surface. Considered as force, the heat comes from whatever neighbouring mass of matter has a greater intensity of this force—or, in other words, a higher temperature—upon its own atoms than that pervading the water. The influent heat, or force, goes to the water-atoms in the same manner as to the atoms of other matter, and the atoms of water are said to be heated. Rumford contended that although water obviously receives heat, it could not *conduct* heat progressively from particle to particle. He insisted also upon the non-conductibility of gases; but while the conducting power of water has been proved by Despretz, that of hydrogen gas is also affirmed upon strong evidence by Professor Magnus. The first atoms of the water to which the heat is attracted will either be warmed or converted into steam, according to the quantity of heat or force with which they become endowed. Upon the inevitable supposition that they receive their force progressively, they can only acquire the vaporous condition when the force of heat centred upon them has completely overcome their attraction for the contiguous particles of water. But with any elevation of temperature, however slight, the atoms, although we need not suppose them to have themselves changed in volume, repel their fellows to an increased distance, and are thus virtually expanded, and thus have correspondingly a less specific gravity. When, therefore, heat is applied to the bottom of a vessel of water, the atoms first expanded are suddenly forced up to the surface by the superior gravity of the colder atoms above them. In rising, the warmed atoms must give off a portion of their heat; and they are likely to arrive at the surface with so little heat that, instead of escaping as steam, they remain there as water. Their ascent from the bottom leaves fresh atoms exposed to the source of heat, and these are in turn warmed, expanded, and forced up as before. It is sometimes said that the atoms on rising "give off their steam," and immediately descend. This explanation is inconsistent with the condition of the atoms left at the bottom; and the descent of atoms already at the surface must be gradual, and at the rate at which still warmer atoms are forced above them. The successive positions of a vertical series of atoms, as the lowermost are successively heated and forced to the surface, may be understood as follows, representing say seven atoms by letters, the atom A being at the top and G at the bottom before the application of heat:—

A	G	F	E	D	C	B	A
B	A	G	F	E	D	C	B
C	B	A	G	F	E	D	C
D	C	B	A	G	F	E	D
E	D	C	B	A	G	F	E
F	E	D	C	B	A	G	F
G	F	E	D	C	B	A	G

This series of movements applies merely to the heating of the water under pressure. It represents the transmission of heat by the process known as convection. At the same time a gradual conduction of heat upwards will be going on in the vertical series of atoms, in the same manner as if they were part of a solid body, and thus having no motion among themselves. This conduction upwards is received by the upper atoms not so much as an increase in their temperature as an increase in the total quantity of heat surrounding them. In this way the upper atom in a vertical series becomes further separated from that next below it than any other pair still lower down are separated. The upper atom, on receiving a sufficient quantity of heat, will thus be detached from the surface of the water, and rise as steam of a temperature depending upon the pressure. In this way a certain quantity of steam will be formed, and it will rise from the surface of the water almost immediately upon the application of heat at the bottom. Much the larger part of the heat, however, will go to warming the water itself. Until a certain intensity of heat is reached, the steam escaping at the surface will not have sufficient force to press the air bodily from over the water, but will insinuate itself among the particles of the air in a manner referable to the well-known action of gaseous absorption. It has been already said that the steam which rises from water while in course of heating in the open air is commonly called "vapour;" but it is none the less true steam, although of but low temperature and elasticity. It diffuses itself in air in precisely the same manner as it would do in a vacuum; and although unavailable in practice as a motive power, it is theoretically capable of giving out mechanical work in proportion to its total contained heat.

In order to rise bodily, or as a piston, so to speak, against air at the ordinary pressure (say 30 inches of mercury), steam must of necessity have an elastic force greater than that of the atmosphere. This force has its corresponding temperature—viz., 212 degrees of Fahrenheit's thermometer. And when water is once heated to 212 degrees under the ordinary pressure of the atmosphere, the steam rising from the bottom no longer bestows its own heat upon the water through which it rises, but arrives with its full force and escapes at the surface. The bodily agitation of the water, which attends the rise and escape of steam of a pressure equal to that upon the water, is termed ebullition, or boiling. In this case a vertical series of atoms, represented, as before, by letters, would have the following successive positions, until the last was evaporated:—

A						
B	A					
C	B	A				
D	C	B	A			
E	D	C	B	A		
F	E	D	C	B	A	
G	F	E	D	C	B	A

The atom A, at the last, taking its modicum of heat, and rising as steam. This successive arrangement of atoms is that which may be supposed to attend the exclusive application of heat at the bottom. If the heat be applied exclusively at the top, the successive conditions of the vertical series of atoms would be thus:—

A						
B	B					
C	C	C				
D	D	D	D			
E	E	E	E	E		
F	F	F	F	F	F	
G	G	G	G	G	G	G

The atom G finally receiving its modicum of heat, and rising as steam.

As the pressure upon boiling water is increased, a higher temperature or intensity of heat-force is requisite in order to overcome this pressure, and thus raise the water into steam. The resistance opposed to the separation of the particles of the water remains the same as before, but a greater resistance has to be overcome by the steam in leaving the water. This not only requires a higher temperature but a greater total quantity of heat, and thus it is that high-pressure steam contains absolutely more heat than low-pressure steam—a truth first definitively established by the experiments of M. Regnault. The determination of the exact pressure and density of steam of different temperatures—a matter of great consequence in calculating the duty of steam engines—has involved many and various experimental researches. The results deduced from those of M. Regnault are now accepted as the nearest to truth. These apply to steam in its ordinary condition, or when accumulated over the body of water from which it has been generated. In this condition steam is said to be saturated, because it has taken up and vaporized the full quantity of water of which its own temperature will admit. Steam of which the density is less than that of saturated steam of the same temperature, and which, therefore, is capable of taking up a further quantity of water, is called superheated steam. All the tables given in the books, of the properties of steam, apply to steam in the saturated state only. These tables, of which one of the best will be given, are usually calculated for pressures increasing progressively by 1 lb., by interpolating between a greater or less number of experimental results. The interpolated or calculated results depend for their value upon an ascertained law; and such is the confidence now felt in our knowledge of the laws which govern the properties of steam, and the expansion of all gaseous bodies, that it is usual to attribute any irregularities, in the comparison of the experimental and calculated results, to slight defects in the experimental apparatus, or to errors of observation.

The law upon which our knowledge of the relations of the density, elasticity, and temperature of steam most depends is, that all gaseous bodies expand equally for equal additions of temperature, and that the progressive rate of expansion is equal for equal increments of temperature. Thus it is not only found that in heating air from 32 degrees it expands $\frac{1}{491}$ of its original volume for each Fahrenheit degree of heat applied, but all other permanent gases expand in the same ratio. Upon the supposition that steam is a perfect gas, the ordinary tables of its temperature, elasticity, and density are calculated; but it has been lately shown that there is some deviation from the law of perfect gaseity, at temperatures greater than 212 degrees. The relations of the properties of steam, as now generally accepted in science, are expressed in the following table:—

PROPERTIES OF SATURATED STEAM.

Total Pressure per sq. inch, measured from a Vacuum.	Pressure above the Atmosphere.	Sensible Temperature in Fahrenheit degrees.	Total Heat in Degrees from Zero of Fahrenheit.	Weight of One Cubic Foot of Steam.	Relative Volume of the Steam compared with the Water from which it was raised.
Lb.	Lb.	Deg.	Deg.	Lb.	
1	...	102·1	1144·5	·0030	20582
2	...	126·3	1151·7	·0058	10721
3	...	141·6	1156·6	·0085	7322
4	...	153·1	1160·1	·0112	5583
5	...	162·3	1162·9	·0138	4527
6	...	170·2	1165·3	·0163	3813
7	...	176·9	1167·3	·0189	3298
8	...	182·9	1169·2	·0214	2909
9	...	188·3	1170·8	·0239	2604
10	...	193·3	1172·3	·0264	2358
11	...	197·8	1173·7	·0289	2157
12	...	202·0	1175·0	·0314	1986
13	...	205·9	1176·2	·0338	1842
14	...	209·6	1177·3	·0362	1720
14·7	0·	212·0	1178·1	·0380	1642
15	·3	213·1	1178·4	·0387	1610
16	1·3	216·3	1179·4	·0411	1515
17	2·3	219·6	1180·3	·0435	1431
18	3·3	222·4	1181·2	·0459	1357
19	4·3	225·3	1182·1	·0483	1290
20	5·3	228·0	1182·9	·0507	1229
21	6·3	230·6	1183·7	·0531	1174
22	7·3	233·1	1184·5	·0555	1123
23	8·3	235·5	1185·2	·0580	1075
24	9·3	237·8	1185·9	·0601	1036
25	10·3	240·1	1186·6	·0625	996
26	11·3	242·3	1187·3	·0650	958
27	12·3	244·4	1187·8	·0673	926
28	13·3	246·4	1188·4	·0696	895
29	14·3	248·4	1189·1	·0719	866
30	15·3	250·4	1189·8	·0743	838
31	16·3	252·2	1190·4	·0766	813
32	17·3	254·1	1190·9	·0789	789
33	18·3	255·9	1191·5	·0812	767
34	19·3	257·6	1192·0	·0835	746
35	20·3	259·3	1192·5	·0858	726

PROPERTIES OF SATURATED STEAM—*Continued.*

Total Pressure per sq. inch, measured from a Vacuum.	Pressure above the Atmosphere.	Sensible Temperature in Fahrenheit degrees.	Total Heat in Degrees from Zero of Fahrenheit.	Weight of One Cubic Foot of Steam.	Relative Volume of the Steam compared with the Water from which it was raised.
Lb.	Lb.	Deg.	Deg.	Lb.	
36	21·3	260·9	1193·0	·0881	707
37	22·3	262·6	1193·5	·0905	688
38	23·3	264·2	1194·0	·0929	671
39	24·3	265·8	1194·5	·0952	655
40	25·3	267·3	1194·9	·0974	640
41	26·3	268·7	1195·4	·0996	625
42	27·3	270·2	1195·8	·1020	611
43	28·3	271·6	1196·2	·1042	598
44	29·3	273·0	1196·6	·1065	585
45	30·3	274·4	1197·1	·1089	572
46	31·3	275·8	1197·5	·1111	561
47	32·3	277·1	1197·9	·1133	550
48	33·3	278·4	1198·3	·1156	539
49	34·3	279·7	1198·7	·1179	529
50	35·3	281·0	1199·1	·1202	518
51	36·3	282·3	1199·5	·1224	509
52	37·3	283·5	1199·9	·1246	500
53	38·3	284·7	1200·3	·1269	491
54	39·3	285·9	1200·6	·1291	482
55	40·3	287·1	1201·0	·1314	474
56	41·3	288·2	1201·3	·1336	466
57	42·3	289·3	1201·7	·1364	458
58	43·3	290·4	1202·0	·1380	451
59	44·3	291·6	1202·4	·1403	444
60	45·3	292·7	1202·7	·1425	437
61	46·3	293·8	1203·1	·1447	430
62	47·3	294·8	1203·4	·1469	424
63	48·3	295·9	1203·7	·1493	417
64	49·3	296·9	1204·0	·1516	411
65	50·3	298·0	1204·3	·1538	405
66	51·3	299·0	1204·6	·1560	399
67	52·3	300·0	1204·9	·1583	393
68	53·3	300·9	1205·2	·1605	388
69	54·3	301·9	1205·5	·1627	383
70	55·3	302·9	1205·8	·1648	378
71	56·3	303·9	1206·1	·1670	373
72	57·3	304·8	1206·3	·1692	368
73	58·3	305·7	1206·6	·1714	363
74	59·3	306·6	1206·9	·1736	359
75	60·3	307·5	1207·2	·1759	353
76	61·3	308·4	1207·4	·1782	349
77	62·3	309·3	1207·7	·1804	345
78	63·3	310·2	1208·0	·1826	341
79	64·3	311·1	1208·3	·1848	337
80	65·3	312·0	1208·5	·1869	333
81	66·3	312·8	1208·8	·1891	329
82	67·3	313·6	1209·1	·1913	325
83	68·3	314·5	1209·4	·1935	321
84	69·3	315·3	1209·6	·1957	318
85	70·3	316·1	1209·9	·1980	314
86	71·3	316·9	1210·1	·2002	311
87	72·3	317·8	1210·4	·2024	308
88	73·3	318·6	1210·6	·2044	305
89	74·3	319·4	1210·9	·2067	301
90	75·3	320·2	1211·1	·2089	298
91	76·3	321·0	1211·3	·2111	295
92	77·3	321·7	1211·5	·2133	292
93	78·3	322·5	1211·8	·2155	289
94	79·3	323·3	1212·0	·2176	286
95	80·3	324·1	1212·3	·2198	283
96	81·3	324·8	1212·5	·2219	281
97	82·3	325·6	1212·8	·2241	278
98	83·3	326·3	1213·0	·2263	275
99	84·3	327·1	1213·2	·2285	272
100	85·3	327·9	1213·4	·2307	270
101	86·3	328·5	1213·6	·2329	267
102	87·3	329·1	1213·8	·2351	265
103	88·3	329·9	1214·0	·2373	262
104	89·3	330·6	1214·2	·2393	260
105	90·3	331·3	1214·4	·2414	257
106	91·3	331·9	1214·6	·2435	255
107	92·3	332·6	1214·8	·2456	253
108	93·3	333·3	1215·0	·2477	251
109	94·3	334·0	1215·3	·2499	249
110	95·3	334·6	1215·5	·2521	247
111	96·3	335·3	1215·7	·2543	245
112	97·3	336·0	1215·9	·2561	243
113	98·3	336·7	1216·1	·2586	241
114	99·3	337·4	1216·3	·2607	239
115	100·3	338·0	1216·5	·2628	237
116	101·3	338·6	1216·7	·2649	235
117	102·3	339·3	1216·9	·2674	233
118	103·3	339·9	1217·1	·2696	231
119	104·3	340·5	1217·3	·2738	229
120	105·3	341·1	1217·4	·2759	227
121	106·3	341·8	1217·6	·2780	225
122	107·3	342·4	1217·8	·2801	224
123	108·3	343·0	1218·0	·2822	222
124	109·3	343·6	1218·2	·2815	221
125	110·3	344·2	1218·4	·2867	219
126	111·3	344·8	1218·6	·2889	217
127	112·3	345·4	1218·8	·2911	215
128	113·3	346·0	1218·9	·2933	214
129	114·3	346·6	1219·1	·2955	212
130	115·3	347·2	1219·3	·2977	211
131	116·3	347·8	1219·5	·2999	209
132	117·3	348·3	1219·6	·3020	208
133	118·3	348·9	1219·8	·3040	206
134	119·3	349·5	1220·0	·3060	205
135	120·3	350·1	1220·2	·3080	203
136	121·3	350·6	1220·3	·3101	202
137	122·3	351·2	1220·5	·3121	200
138	123·3	351·8	1220·7	·3142	199
139	124·3	352·4	1220·9	·3162	198
140	125·3	352·9	1221·0	·3184	197
141	126·3	353·5	1221·2	·3206	195
142	127·3	354·0	1221·4	·3228	194
143	128·3	354·5	1221·6	·3250	193
144	129·3	355·0	1221·7	·3273	192
145	130·3	355·6	1221·9	·3294	190
146	131·3	356·1	1222·0	·3315	189
147	132·3	356·7	1222·2	·3336	188
148	133·3	357·2	1222·3	·3357	187
149	134·3	357·8	1222·5	·3377	186
150	135·3	358·3	1222·7	·3397	184
155	140·3	361·0	1223·5	·3500	179
160	145·3	363·4	1224·2	·3607	174
165	150·3	366·0	1224·9	·3714	169
170	155·3	368·2	1225·7	·3821	164
175	160·3	370·8	1226·4	·3928	159
180	165·3	372·9	1227·1	·4035	155
185	170·3	375·3	1227·8	·4142	151
190	175·3	377·5	1228·5	·4250	148
195	180·3	379·7	1229·2	·4357	144
200	185·3	381·7	1229·8	·4464	141
210	195·3	386·0	1231·1	·4668	135
220	205·3	389·9	1232·3	·4872	129
230	215·3	393·8	1233·5	·5072	123
240	225·3	397·5	1234·6	·5270	119
250	235·3	401·1	1235·7	·5471	114
260	245·3	404·5	1236·8	·5670	110
270	255·3	407·9	1237·8	·5871	106
280	265·3	411·2	1238·8	·6070	102
290	275·3	414·4	1239·8	·6268	99
300	285·3	417·5	1240·7	·6469	96

The practical value of this table will be understood as follows:—It being supposed that a locomotive engine is worked with saturated steam, and without expansion, let the working pressure above the atmosphere be 120·3 lbs. per square inch, and suppose 8 lbs. of water to be evaporated per lb. of coal,—at what rate of consumption per indicated horse-power per hour ought the engine to work, supposing one-tenth of the steam to be lost in the steam ports and in clearance at the ends of the cylinders, and supposing the back pressure to average 2 lbs? The result, it will be understood, does not depend upon the dimensions of the engine itself. We first find the volume of water (8 lbs.) vaporized by 1 lb. of coal. This is 0·1282 of 1 cubic foot, and as the relative volume of steam of 120·3 lbs. pressure is 203, then 8 lbs. of water will form 203 × 0·1282 = 26 cubic feet of steam of that pressure. Deducting the quantity of steam estimated to be wasted in steam ports and clearance, 23·4 cubic feet remain; and the effect of this will be that exerted in forcing a piston 1 foot square through a distance of 23·4 feet, with the full pressure of the steam, less 2 lbs. per square inch estimated as back pressure. A piston of 1 square foot, or 144 square inches, pressed with steam of 120·3—2 = 118·3 lbs. per square inch, is loaded with 144 × 118·3 = 17,035·2 lbs., and this pressure exerted through a distance of 23·4 feet gives 398,624 foot-pounds of effective mechanical work performed by 1 lb. of coal. A horse-power is the amount of work expended in lifting 33,000 lbs. 1 foot high in 1 minute, and, therefore, sixty times as much, or 1,980,000 foot-pounds in 1 hour. Therefore the engine, worked under the conditions above considered, would require $\frac{1,980,000}{398,624} = 4·97$ lbs. of coal per indicated horse-power per hour.

Another application of the column of relative volumes of steam in the table will be seen in the following case:—Suppose an engine to have 16-inch cylinders, 24-inch stroke, and 6-feet driving wheels; suppose the port space and clearance at each end of each cylinder to be 288 cubic inches, or ⅙ cubic foot; the steam to be admitted upon the pistons at a pressure of 90·3 lbs. per square inch, and cut off at one-third stroke: What quantity of water will be required in running 1 mile, and what will be the consumption of coal per mile, allowing 1 lb. of coal to 8 lbs. of water?

A 16-inch cylinder has 201 square inches of piston area. The steam being cut off at one-third stroke, or 8 inches, 1,608 cubic inches of steam, or, including that in the port and end clearance, 1,896 cubic inches are admitted at each stroke. Four times this quantity, or 4·39 cubic feet, are admitted to the pair of cylinders at each revolution of the driving wheels, and as a 6-feet wheel makes 280 revolutions in rolling 1 mile, 1,229·2 cubic feet of steam are used per mile. The pressure of steam in the cylinders, at the moment of cutting off the admission, being 90·3 lbs. per square inch, we find by the table that it occupies 257 times the volume of the water from which it was generated. Hence $\frac{1229·2}{257} = 4·78$ cubic feet, or 298¼ lbs. of water, are required per mile; and, with an evaporation of 8 lbs. of water per lb. of coal, the mileage consumption of the latter is 37¼ lbs.

The column of total heat in the table contains the information requisite for many useful calculations as to heating water and condensing steam. Thus, it is known from experiment that 1 lb. of good coal burnt on the grate of a locomotive engine will give an evaporation as high as 8 lbs. of water, at 60 degrees, into steam of 120 lbs. As the total heat of steam of this pressure is 1,220 degrees above the zero of Fahrenheit's scale, each lb. of water is in this case heated 1,220—60 = 1,160 degrees, and the lb. of coal, therefore, has usefully given out 1,160 × 8 = 9,280 units of heat. Knowing this, it is easy to see that the same lb. of coal, burnt in the same manner, would have warmed 100 lbs. of water 92·8 degrees, or 1,000 lbs. of water rather more than 9¼ degrees. The heating power of different coals is, of course, very variable, and the same coal will give different results when burnt in different furnaces, or at different rates of combustion; but the average given will cover a wide range of practice.

If we have steam of 30·3 lbs. pressure above the atmosphere to condense with water of 50 degrees temperature, and if we allow the water to be heated only to 100 degrees, the column of total heat enables us to readily find what quantity of condensing water is required. Steam of 30·3 lbs. has a total heat of 1,197·1 degrees, and in condensing into water of 100 degrees it will part with 1,097 degrees of heat. If it is to heat the water from 50 degrees to 100 degrees only—or, in all, 50 degrees—then the quantity of condensing water should be $\frac{1097}{50}$ = nearly twenty-two times the weight of the steam condensed; or, to put it differently, the cold water pump of a condensing engine would, in this case, require to throw twenty-two times as much water as the feed pump.

It will be borne in mind that the total heat, as also the sensible temperatures given in the table, are measured from the arbitrary zero of Fahrenheit's thermometer. The results of the above calculations would be the same if, instead of zero, the freezing point were taken as the foot of the thermometric scale. The total heat of the steam in the last example would then be 1,165 degrees, and the condensing water would be warmed from 18 degrees to 68 degrees above freezing. The steam would be cooled to 68 degrees above freezing, and would thus lose 1,097 degrees, and it would heat twenty-two times its own weight of water as before.

There is, however, an imaginary absolute zero of temperature often employed in calculating the relations of the properties of gaseous bodies, and which is assumed from the fact that a perfect gas is expanded $\frac{1}{493}$ of its own volume upon the application of each Fahrenheit degree of temperature above the freezing point. From this fact it is inferred that a perfect gas would entirely lose its elasticity, and have the volume due to the solid state, at 493 degrees below the freezing point, or 493 — 32 = 461 degrees below the zero of Fahrenheit's thermometer. The imaginary zero is fixed upon the idea that the expansion or contraction from the freezing point is exactly $\frac{1}{493}$ of the volume for each degree, no matter how great the range of temperature may be. This assumption, it is known, pretty closely covers a wide range of temperature, and it is of value in calculating such tables as that already given.

Beyond the power calculable from the table on page 111, steam gives out an additional power by its own expansion after its admission into the cylinder has been shut off during the stroke of the piston. It has been common to assume, upon the law propounded by Boyle, that the force of steam is inversely as the space occupied. Thus it is commonly estimated that steam of 100 lbs. pressure will, when expanded into twice its original volume, press with a force of 50 lbs. In this case, as in all calculations relating to the power of steam, it must not be forgotten that the original and final pressures are to be measured from a vacuum, and not, as shown by a steam gauge, from the pressure of the atmosphere. Thus, upon Boyle's law (otherwise known as Marriotte's), steam of 85 lbs. above the atmosphere would, by expanding into a double volume, acquire a pressure of 35 lbs., the total pressures before and after expansion being 85 + 15 = 100 lbs. and 35 + 15 = 50 lbs. If steam be cut off at half stroke, it would, in the same way, have three-fourths its original pressure at two-thirds' stroke, two-thirds at three-fourths' stroke, four-sevenths at seven-eighths' stroke, and one-half its original pressure at the end of the stroke, or full stroke. For at two-thirds' stroke the volume of the steam is one-third greater than at half-stroke, at three-fourths' stroke it is one-half greater, at seven-eighths' stroke three-fourths greater, and at full stroke twice as great. In this way the average pressures of steam of a given original pressure may be found at any number of points in the stroke after shutting off the steam. Thus, suppose a 24-inch stroke, and steam of 100 lbs. total pressure, as measured from a vacuum, to be cut off at half stroke, the mean pressure at each inch of the stroke, after cutting off, would be as follows:—

Inches of Stroke. Steam cut off at 12th inch.	Volume of Steam as compared with Original Volume.	Pressure of Steam at each inch of Stroke after 12th inch.
13th,	Thirteen-twelfths,	1,200 ÷ 13 = 92·3 lbs.
14th,	Fourteen-twelfths,	1,200 ÷ 14 = 85·7 ,,
15th,	Fifteen-twelfths,	1,200 ÷ 15 = 80·0 ,,
16th,	Sixteen-twelfths,	1,200 ÷ 16 = 75·0 ,,
17th,	Seventeen-twelfths,	1,200 ÷ 17 = 70·6 ,,
18th,	Eighteen-twelfths,	1,200 ÷ 18 = 66·7 ,,
19th,	Nineteen-twelfths,	1,200 ÷ 19 = 63·1 ,,
20th,	Twenty-twelfths,	1,200 ÷ 20 = 60·0 ,,
21st,	Twenty-one-twelfths,	1,200 ÷ 21 = 57·1 ,,
22d,	Twenty-two-twelfths,	1,200 ÷ 22 = 54·5 ,,
23d,	Twenty-three-twelfths,	1,200 ÷ 23 = 52·2 ,,
24th,	Twenty-four-twelfths,	1,200 ÷ 24 = 50·0 ,,
		12)807·2 lbs.
	Average pressure during expansion, .	67·2 lbs.

And as the steam pressed with a force of 100 lbs. up to half stroke, where it was cut off, we have 100 + 67·2 = 167·2 to be divided by two, for the mean pressure, = 83·6 lbs. throughout the whole stroke; whereas, but for expansion the same weight of steam would have given an average pressure of but 50 lbs. throughout the whole stroke, the effect of the steam being thus increased 67 per cent. by expansion.

It is to be understood, in computations of this kind, that although the steam which fills the ports and end clearance in the cylinder is wholly lost when the piston is worked at full pressure during its whole stroke, useful pressure is obtained from this steam during expansion. The port space and end clearance sometimes amount to nearly or quite one-tenth of the full capacity of the cylinder, so that, in cutting off at one-tenth of the stroke, the amount of steam expanded would be really equal to one-fifth the capacity of the cylinder, the final pressure being theoretically one-fifth, instead of one-tenth, of the original or "initial" pressure. Disregarding for a moment the effect of port space and clearance, the average pressure, in the case considered in the preceding paragraph, would be 83·6—14·7 = 68·9 lbs. above the atmosphere; and thus, if no condensing apparatus be employed, this will be the pressure to compare with 85 lbs., the full pressure, in estimating the gain by expansion. Here the gain by expansion will be rather less than 60 per cent. If there be any back pressure above the atmosphere, the gain by expansion in this case will be still less.

The illustration already given of the expansive effect of steam is intended rather to show the elementary mode of estimating it than as a formula for the engineer. The effect of expansive working is more quickly and accurately found with the aid of a table of hyperbolic logarithms, as follows:—

HYPERBOLIC LOGARITHMS.

No.	Hyp. Log.	No.	Hyp. Log.	No.	Hyp. Log.	No.	Hyp. Log.
1·05	·049	2·3	·833	4·1	1·411	6·6	1·887
1·1	·095	2·35	·854	4·2	1·435	6·7	1·902
1·15	·140	2·4	·875	4·3	1·459	6·8	1·917
1·2	·182	2·45	·896	4·4	1·482	6·9	1·931
1·25	·223	2·5	·916	4·5	1·504	7·0	1·946
1·3	·292	2·55	·936	4·6	1·526	7·1	1·960
1·35	·300	2·6	·956	4·7	1·548	7·2	1·974
1·4	·336	2·65	·975	4·8	1·569	7·3	1·988
1·45	·372	2·7	·993	4·9	1·589	7·4	2·001
1·5	·405	2·75	1·012	5·0	1·609	7·5	2·015
1·55	·438	2·8	1·030	5·1	1·629	7·6	2·028
1·6	·470	2·85	1·047	5·2	1·649	7·7	2·041
1·65	·500	2·9	1·065	5·3	1·668	7·8	2·054
1·7	·531	2·95	1·082	5·4	1·686	7·9	2·067
1·75	·560	3·0	1·099	5·5	1·705	8·0	2·079
1·8	·588	3·1	1·131	5·6	1·723	8·1	2·092
1·85	·615	3·2	1·163	5·7	1·745	8·2	2·104
1·9	·642	3·3	1·194	5·8	1·758	8·3	2·116
1·95	·668	3·4	1·224	5·9	1·775	8·4	2·128
2·0	·693	3·5	1·253	6·0	1·792	8·5	2·140
2·05	·718	3·6	1·281	6·1	1·808	8·6	2·152
2·1	·742	3·7	1·308	6·2	1·824	8·7	2·163
2·15	·765	3·8	1·335	6·3	1·841	8·8	2·175
2·2	·788	3·9	1·361	6·4	1·856	8·9	2·186
2·25	·811	4·0	1·386	6·5	1·872	9·0	2·197

This table is employed in the following manner in estimating the gain by expansive working:—The whole stroke of the piston is divided by that portion of the stroke during which steam is admitted by the valve, and the hyperbolic logarithm of the quotient is found in the table. Thus, if steam be admitted for 8 inches of a 24-inch stroke, the quotient, by dividing the latter by the former number, is 3, and the hyperbolic logarithm of this is 1·099. But to the logarithm the whole number 1 is to be added, and the sum, compared with unity, then expresses the ratio of the gain by expansion. Thus, in the case first considered, the sum obtained by adding 1 to the hyperbolic logarithm of 3 is 2·099; and this, compared with 1, shows the effect of the steam to be rather more than doubled by cutting off at one-third stroke and expanding. If the stroke be 20 inches, and the steam be cut off at 2½ inches, the quotient on division is 8, the hyperbolic logarithm of which is 2·079, and, adding 1 to this, it becomes 3·079,—showing that, as compared with 1, the effect by expansion is rather more than three-fold that of the steam worked at full stroke. Although not given in the above table, the hyperbolic logarithm of 10 is 2·303; of 100, 4·605; of 1,000, 6·908; and of 10,000, the hyperbolic logarithm is 9·210. Thus, in theory, the effect by expanding steam from the one ten-thousandth part of the stroke, or ten thousand-fold, would be ten and a quarter times nearly that of the same steam worked at full stroke. In such suppositions, a condensing engine, working with a perfect vacuum, must be also supposed, as steam of any practicable pressure would, even when moderately expanded, soon fall below the pressure of the atmosphere; in which case expansion, so far from being attended with gain, would involve a loss, from the fact that the pressure on the driving side of the piston would be less than that of the atmospheric resistance on the exhausting side. The atmospheric resistance, and other back pressure, if any, against the pistons of non-condensing engines, must always be considered in dealing with the question of expansive working. This will appear as follows:—Let steam of 100 lbs. pressure above the atmosphere be expanded eight-fold in each of two engines, one condensing, the other non-condensing,—the total pressure, down to a perfect vacuum, may be taken as 115 lbs. And it is evident, upon the usual assumption that the volume of steam is inversely as its density, that the same weight of steam is used, and, consequently, the same expenditure of coal incurred, whether steam of this pressure be cut off at one-eighth of the stroke, or whether steam of one-eighth this pressure be worked at full stroke. But if steam of $\frac{115}{8}$ = 14⅜ lbs. total pressure be worked at full stroke, it can only be worked in a condensing engine, its pressure being already below that of the atmosphere. In cutting off steam of 115 lbs. total pressure at one-eighth stroke, its effect, as compared with the same weight of steam worked at full stroke, will be (hyperbolic logarithm of $8 = 2{\cdot}079$, and this with the addition of unity $= 3{\cdot}079$) rather more than three times as much, the average pressure throughout the stroke being $14{\cdot}375 \times 3{\cdot}079 = 44\frac{1}{4}$ lbs. But in a non-condensing engine, with the same degree of expansion, the steam of 115 lbs. total pressure would expand below the atmosphere before the end of the stroke was reached. Apart from this, the average driving pressure would be $44\frac{1}{4} - 14{\cdot}7 = 29\frac{1}{2}$ lbs. only. To use the same weight of steam throughout the full stroke, the piston should have but one-eighth the area of those of the engines just considered. Here the average effective pressure would be 100 lbs.—equal in effect to 12½ lbs. above the atmosphere on a piston of the full size first considered. The gain by expansion would be estimated by comparing the pressures of 29½ lbs. and 12½ lbs., instead of 44¼ lbs. and 14⅜ lbs., as in the condensing engine, the advantage in the latter case being at once apparent.

As has been already observed, the effect of expansive working is not to be estimated merely from a consideration of the fraction of the stroke of the piston corresponding to the admission of steam. The steam in the port space and in the clearance between the piston and cylinder cover is practically lost so long as steam is being admitted to the cylinder; but at the moment of cutting off, this lost steam at once shares in the expansive duty, and contributes to the effect produced. It is not uncommon to hear of locomotives cutting off at one-eighth stroke, although the highest pressure of steam reaching the piston may not exceed 90 lbs., or 105 lbs. if we include the pressure of the atmosphere. Here an eight-fold expansion would give a terminal pressure nearly 2 lbs. below the atmosphere; whereas, by the expansion of steam from the port and end clearance, the ratio of expansion may be only five-fold, and the terminal pressure say 6 lbs. above the atmosphere. The piston, nevertheless, has had the full pressure of the steam for but one-eighth, and not for one-fifth its stroke.

The theoretical advantages of expansion have been understood since the time of Watt; yet nothing is more common than to hear of high-pressure steam and high expansive working as instances of modern progress in steam engineering. Woolf, in 1806, proposed extravagant rates of expansion, preferring steam of 50 lbs. pressure. The American, Evans, had still earlier proposed, and, indeed, employed steam of a much higher pressure; and there was hardly a limit to the pressures afterwards proposed, and, indeed, worked, by Perkins, Alban, and Craddock, the first and the last-named engineers having carried expansion to a great extent. These facts are mentioned here in order that the reader may be at once guarded against an assumption which is very prevalent, that high expansive working is attended with unmixed good. The history, however, of the repeated attempts made to carry it into practice shows that the limit of economical expansion is soon reached. If we suppose a condensing engine to work with full-pressure steam during the whole stroke, with an expenditure of 5 lbs. of coal per indicated horse-power per hour, the theoretical consumption of coal would be as follows, for different rates of expansion:—

Point of Cut-off.	Lbs. Coal per Hourly Horse-power.	Comparative Capacity of Cylinder for a given power.
Full stroke, . .	5 lbs. .	1·00
One-third, . .	2½ " .	1·50
One-eighth, . .	1⅔ " .	2·66
One-twentieth, .	1¼ " .	5·00
One-fifty-fifth, .	1 " .	11·00
One-thousandth, .	⅝ " .	112·00
One-ten-thousandth,	½ " .	980·00

The last column shows what would be the necessary relative capacities of the cylinder, supposing the same pressure of steam to be employed at each rate of expansion. At the higher rates, however, a higher pressure of steam would be desirable,—especially with a safe and practicable high-pressure boiler, the want of which has, until recently, greatly retarded the introduction of high-pressure steam.

It will be seen that, although theoretically the consumption of fuel is diminished one-half by cutting off at one-third stroke as compared with full-stroke, a further diminution of only one-sixth of the original consumption is effected even by an eight-fold rate of expansion; while a further saving of but one-twelfth of the original consumption would result from a twenty-fold rate of expansion. While it is not difficult to save even one-half of the fuel burned in working at full stroke, it is virtually impracticable, after this saving has been once effected, to save even one-half of the remaining half; while, at the same time, this subsequent saving is of but half the magnitude or commercial value of that first effected. And in no case could a considerable rate of expansion be effected unless the engine was fitted with condensing apparatus, or unless the working pressure was very high. In order to expand twenty-fold in a non-condensing engine it would be necessary to employ steam of 300 lbs. pressure; for with anything less than this the final pressure would go below the atmosphere, and thus cause a loss.

The objections to high expansive working are many and various. First, either a great pressure of steam must be resorted to, or else a cylinder of great size must be employed. With very high pressures, few boilers can be depended upon, and in any case there is considerable leakage in the packings, and even the high temperature of very high-pressure steam gives trouble. Thus, on the London and North-Western Railway, and, indeed, on several others, the working pressure has been reduced from 150 lbs. to 120 lbs., in consequence of the difficulty experienced at the higher rate with the packings, and in respect also of lubrication. If we choose to refer to steam of 300 lbs. pressure, its temperature is nearly that of the melting point of pure tin, the former being 422 degrees and the latter 426 degrees. If, instead of a very high pressure, a very large cylinder be employed, the condensing apparatus must be very perfect, and in any case the cylinder and working parts must be of great weight and bulk, and there must be a comparatively great surface for the radiation of heat. Again, whether very high-pressure steam or a very large cylinder be employed, the initial blow upon the piston is much above the mean motive force exerted throughout the stroke, and hence the working parts must have a strength much beyond the average stress brought upon them; and hence, also, the motion, unless controlled by either a heavy or a rapidly revolving fly-wheel, must be correspondingly irregular. This irregularity of motion may be corrected to a great extent by first working the steam for say half stroke upon a small piston, and afterwards exhausting upon a larger one, as in the compound, or high and low-pressure engines, originally invented by Hornblower in 1776, and improved by Woolf in 1806, since which time they have been extensively employed. The compound engine involves the adoption, however, of a considerable additional number of parts, and it has not yet been applied in any form to locomotives. Highly expansive engines generally work also under a disproportionate valve friction, the valves working constantly in full-pressure steam. A further and a serious disadvantage is in the cooling of the internal surfaces of the cylinder during great expansion, and the consequent condensation of a large quantity of high-pressure steam, on its admission on the succeeding stroke. Thus if, to obtain great expansion, steam of 300 lbs. pressure be expanded down to 1 lb. above a vacuum, the range of temperature at each stroke will be from 422 degrees to 100 degrees, and so far as the internal surfaces of the cylinders adapt their temperatures to this range—and they must do so—so will be the condensation, to little purpose; for it is true that, after the cylinder is once warmed, at each stroke a certain amount of condensed steam will be re-evaporated towards the end of the stroke.

Not only, from the practical reasons already considered, are the theoretical results inferable from the laws of expansion materially modified, but a new disturbance is introduced by the passage of hot water along with the steam from the boiler to the cylinders. What this amount of water mechanically suspended in the steam may be, is not known, nor can it well be ascertained. It is a matter of some importance, however, and it may cause an apparent gain by expansion, where there is in reality a large loss of heat by the nearly useless removal of heated water from the boiler. Most boilers, if hard worked, are apt to "prime," or to send out spray with the steam; but where priming happens at all, it is almost impossible to define its extent. Thus, a foul boiler will prime, and the loss of water by priming is very often and very carelessly credited to evaporation. But in checking the evaporation of a given boiler, it has been often found that less water disappeared, per pound of fuel burnt, *after* the boiler had been washed out, than before. The apparent difference against a clean boiler could only result from the fact that the priming was less. Although priming has its causes, and although these must be within the reach of investigation, it has been regarded as an obscure malady, and its predisposing causes have not possibly been very carefully sought. With but slight differences of structure or proportions, some boilers are always working water over into the cylinders, while others deliver their steam, so far as accessible evidence goes, perfectly dry. The consideration of these points will arise further on; but it is well to note, at this point, that the question of priming does enter into that of expansive working. Apart from superheating, it is important to deliver the steam in a perfectly dry or gaseous state to the cylinder. It is quite probable that steam-drying apparatus, already employed in a few varieties of locomotives, may be brought still further into use. There are, upon the present view of the case, certain practical difficulties in the way; but these, it is not unlikely, may be overcome.

It is difficult to say exactly where steam passes from the saturated to the superheated state. This difficulty may be understood from Professor Rankine's statement of the difficulty of determining when steam passes from the cloudy or misty state into that of ordinary saturation. He says,—"A vapour near the point of liquefaction has the power of retaining suspended in it a portion of its liquid in the state of cloud or mist; and if, in experiments on the density and expansion of steam, glass vessels are used, in order to show when the steam is free from cloud, a new cause of uncertainty is introduced by the fact that the attraction between glass and water is sufficient to retain in the liquid state, and in contact with the glass, a film of water at a temperature at which, but for the attraction of the glass, it would be in the state of steam." Thus, in applying heat to steam already removed from water, it is probable that the first effect, so far from actually superheating the whole mass to a temperature above that normally due to its pressure, is merely to vaporize particles of mist or cloud mechanically suspended within it. In 1852, Mr. C. W. Siemens communicated to the Institution of Mechanical Engineers the results of a series of experiments, showing an unexpectedly high rate of expansion of steam, for the first few degrees of temperature applied to it after its removal from the water from which it was generated. Mr. Fairbairn has more recently made similar experiments, with much the same result. So long as the increased elastic force, acquired by steam when heated out of contact with water, is due to the conversion of watery particles of mist, the heat of conversion will be the same as for ordinary liquid water of the same temperature. But when all the water has once taken the condition of true steam, then its expansion, under additional heat, proceeds upon a new ratio. For the specific heat of steam, when allowed to expand by heat, under a constant pressure, is but 0·475 that of water; that is, the quantity of heat necessary to raise the temperature of 1 lb. of water 1 degree would raise that of more than 2 lbs. in weight of steam by the same amount. If the steam be maintained at a constant volume, so that its pressure increases by the application of heat, then its specific heat, as compared with that of water, is still less, being but 0·365. And thus, although the rate of expansion of steam for a given number of degrees above the saturation point may be much more rapid than after the steam is once dried, yet the total quantity of heat absorbed in the first case will be much greater than that taken up in the second. It would appear that, could superheating apparatus be maintained, and superheated steam worked, at high temperatures—the steam being kept under constant volume while in course of superheating—1 lb. of coal would effect nearly six and a half times as much work in superheating than in evaporation. For the heat that would impart 1,200 degrees of total temperature to water in the process of evaporation, would impart 3,300 degrees to the same weight of steam under constant volume. And as dry steam, reckoning from a temperature of 32 degrees, would receive an additional pressure of one atmosphere for about every 500 degrees of heat applied, it would acquire a pressure of nearly 7¼ atmospheres on the addition of 3,300 degrees temperature. This estimate excludes the consideration of expansive working, which would increase the calculated gain. A temperature from which great gain may be had by superheating cannot be tolerated, however, in steam engines, and the temperature just assumed—viz., 3,300 degrees—in addition to the original temperature of the steam, is, of course, as a matter of practice, wholly out of the question. But the same gain would result by applying the same quantity of heat (or the heat derived from the combustion of the same weight of fuel) to say ten times the weight of steam, raising it to one-tenth the temperature. Even this, however, say 330 degrees, could not be added to the existing temperature of ordinary steam without the destruction of the packings and the evaporation of all the lubricating material in the cylinders and valve boxes. In practice, the addition of 100 degrees to the temperature of steam, which in its ordinary or saturated state is already at 300 degrees or 350 degrees, is found to be all that can be prudently ventured upon. The action of most superheating apparatus depending upon the waste heat from the boiler is almost of necessity irregular; and hence, as the variation of a few degrees may lead to heating and "cutting" some of the working parts of the engine, very little more than a thorough drying of the steam is now attempted. The use of wet steam is attended with two distinct kinds of loss. The water suspended in it, and having a temperature the same as that of the steam itself, is wholly thrown away in the case of non-expansive engines; and even with expansion but a small return is made, by the evaporation of a small portion of this water within the cylinder, for the heat which it has carried from the boiler. If half of the whole weight of the steam be due to the weight of suspended water, and the steam have a temperature of 350 degrees, about one-fourth more coal is necessary than would be consumed in the production of dry steam doing the same work. Again, after the steam has done its work, its escape from the cylinder is much less rapid when it is in a foggy condition; and thus the back pressure against the piston is increased. If damp steam has any countervailing advantage, it is as a tolerable lubricant, although, as compared with oil or tallow, it is a most expensive one. Engines are known which, during many years of work, have never had oil or tallow admitted within their cylinders or valve boxes.

In considering the action of steam in an engine, it is to be remembered that, as heat and force are mutually convertible, according to the views already set forth in the present chapter, there must be a positive loss of heat in the production of work upon the piston. The mutual convertibility of heat and force comes to this: Whether a body be heated by thermal conduction, or compressed or abraded by mechanical force, it is both heated and energized. By cooling a body, whether by exposure to a colder body or by allowing it to give off work by expansion, it is not only cooled, but is in some proportion deprived of energy. By "energy" is now understood the whole amount of mechanical work in foot-pounds which a body stores up in being heated, or gives off in cooling; and in thus using the term energy, the heating and cooling of gases are supposed to be referred to the absolute zero of temperature, which is 461 degrees below the zero of Fahrenheit's thermometer. A bar of iron by being heated stores up expansive force which it did not before possess. If tightly enclosed on all sides, it will exert this force strongly, or, if water be dropped upon it, its heat will vaporize a portion, the energy or total mechanical work in this vapour being now readily calculable. But in speaking of heating a bar of iron, it is to be borne in mind that it can be heated also by compressive force. By quick hammering, a blacksmith will heat a horse-nail to redness. In doing this he has merely transferred his own muscular force to the iron—and to the hammer; for this, and even the hammer handle and the anvil, will have acquired a part of this force. The heated iron, the hammer, the anvil, and everything which has acquired heat during the process of hammering, is capable, in theory at least, of again giving out all the force which has been impressed upon it. So in cooling. Whether a volume of air or of steam lose heat by conduction to some other substance, or by giving out force by expansion, in either case it loses both repulsive force and heat; and heat and repulsive force are in fact one and the same.

The exact mechanical equivalent of heat-force has within the last twenty years received much attention from men of science. The British thermal unit, or the heat required to raise 1 lb. of water at 39 degrees through 1 degree, must, upon a very scanty reflection, have some measure of mechanical value, as in foot-pounds of work. Rumford saw this in the latter part of the last century, and Seguin (the original patentee of the multitubular boiler in France, as he was also the first engineer to construct wire suspension bridges) approached closely to the mechanical equivalent of heat in 1839.* In 1842 a German philosopher, Mayer, came still nearer, and, taking the testimony of Professor Tyndall, he arrived at his conclusions by a simple and convincing, as it was indeed a beautiful, induction.

Mayer took the rate of expansion of a gas, as air, from the freezing point of water, and under constant pressure, as $\frac{1}{273}$d of its own volume for each degree Centigrade, or $\frac{1}{491}$st for each degree Fahrenheit. He noted, according to Professor Tyndall, the weight of combustible required to double a given volume of air under constant pressure. He also noted the quantity of combustible required to double the pressure of a given weight of air while preserving a constant volume. Where 10 grains or 10 lbs. of a given kind of fuel were used in the latter case, 14·21 grains or 14·21 lbs. of the same kind of fuel were used in the former. Thus it appeared that for every 14·21 grains of fuel burnt in the former case, 4·21 grains were expended in the mechanical work of forcing the heated column of air against the given resistance, which may be represented as a weight lifted. To double the volume of a cubic foot of air at 32 degrees, an atmospheric resistance equal to 144 square inches × 14·7 lbs. per square inch = 2,116·8 lbs. must be raised through one foot. In other words, 2,116·8 foot-pounds of mechanical work must be exerted. A cubic foot of air at 32 degrees weighs 0·080728 lb. avoirdupois; and as the specific heat of air under constant pressure has been found to be 0·238, that of water being 1·0, the amount of heat required to raise the air 491 degrees Fahrenheit would raise that of 0·01921 lb. of water to the same extent, or, conversely, 9·43 lbs. of water 1 degree Fahrenheit. But where 14·21 grains, or 14·21 lbs. of fuel are required to double the volume of a given weight of air under constant pressure, only 10 grains, 10 oz., or 10 lbs., as the case may be, are required to double the pressure of the same weight of air under constant volume, although the final temperature of the air will be the same in both cases. Thus, if the air be heated from 32 degrees, its final temperature in both cases would be 32 + 491 = 523 degrees. Thus, if the heat required to double the volume of 1 cubic foot of air at 32 be equivalent to that absorbed by 9·43 lbs. of water in rising through 1 degree, that required to double the pressure of the same quantity of air without expansion would be but $\frac{10}{14\cdot21}$ as much, equal to raising 6·64 lbs. of water through 1 degree. Now, the mechanical work overcome in doubling the volume of the cubic foot of air was 2,116·8 foot-pounds; and this must therefore have been effected by the combustion of as much fuel as would have warmed 9·43 − 6·67 = 2·76 lbs. of water 1 degree. Consequently, the heat which would warm 1 lb. of water 1 degree Fahrenheit would raise $\frac{2116\cdot8}{2\cdot76}$ = 767 lbs. 1 foot high. Professor Tyndall gives slightly different quantities; and he takes the pressure of the atmosphere as 15 lbs. per square inch, thus giving Mayer's mechanical equivalent of the thermal unit as 771·4 foot-pounds.

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* Vide *Sur L'Influence des Chemins de Fer*, par Seguin, 1839.

There is room for grave doubt whether Mayer as early as 1842 had the exact specific heats of air, under constant pressure and under constant volume, to work with; but, however this may be, the above is one of the most satisfactory demonstrations, upon accepted data, of the mechanical equivalent of heat—a matter which has come to exercise an important bearing upon almost every question of the economical working, or, rather, the dynamic efficiency of "heat engines," including those worked by steam, hot air, or gas. In 1843, Dr. Joule, of Manchester, made a series of experiments upon the heat imparted to water by churning it with a measured quantity of mechanical power; and although the results derived from experiments of this kind were at first viewed with suspicion, accumulating evidence has led to the general acceptance of Dr. Joule's equivalent as 772 foot-pounds of mechanical power, convertible into the heat represented by heating 1 lb. of water at 39 degrees through 1 degree Fahrenheit, or, conversely, in cooling it by the same amount. This equivalent was experimentally obtained with water, not at 39 degrees, but at 55 degrees to 60 degrees; but it is now generally applied to water at its maximum density.

With this equivalent ascertained, we may arrive at the total energy, or the total possible mechanical work, of which a given weight of steam is capable. A pound of steam, of say 120 lbs. pressure above the atmosphere, is virtually 1 lb. of water heated 1,681 degrees above the absolute zero of a perfect gas thermometer, 1,220 degrees above Fahrenheit's zero, 1,188 degrees above the freezing point, or 1,118 degrees above the sensible temperature of steam of 1 lb. absolute pressure per square inch, the lowest pressure to which a condensing expansive engine could be expected to work. Either of these total temperatures multiplied by 772 will give the energy, in foot-pounds, theoretically due to the steam when worked down say into water of the corresponding temperature. And it must be remembered that, as a gas, to which steam is in this case necessarily compared, it would, upon the accepted law of expansion, only lose its elasticity at a temperature 493 degrees below freezing. If we work down to 102 degrees, the temperature of water from which steam of 1 lb. total pressure would escape, we shall have an energy, for 1 lb. in weight of steam, of 863,096 foot-pounds; and if 10 lbs. of steam be evaporated from 102 degrees, by 1 lb. of coal, giving 8,630,960 foot-pounds per pound of coal, an engine working up the full power of the steam would require but $\frac{1980000}{8630960}$ = 0·23 lbs., or less than 4 oz. of coal per indicated horse-power per hour, an hourly horse-power being 33,000 × 60 = 1,980,000 foot-pounds. To obtain such a result, the steam must, in the very act of doing work, be reduced to 1 lb. of water at 102 degrees. This, however, is quite a theoretical deduction, and nothing like it could, with our present knowledge, be approached in practice, especially as, in expanding, the steam is constantly losing heat and liquefying in the very act of doing work, and thus losing pressure, apart from the loss due to the apparent enlargement of volume. Thus 1 lb. of steam, of a pressure of 120·3 lbs. above the atmosphere, occupies 3·27 cubic feet of space. Suppose it to be expanded down to steam of 1 lb. absolute pressure; if the pressure were inversely as the volume, this would require a 135-fold rate of expansion. And as the hyperbolic logarithm of 135 is 4·75 nearly, the pound of steam thus expanded should effect say 5·75 times as much work as if worked without expansion. Now, without expansion the steam would move a piston of 1 square foot through a stroke of 3·27 feet, with a force (against a vacuum) of 135 lbs. per square inch, or 144 × 135 × 3·27 = 63,569 foot-pounds; and 5·75 times this, or 365,521 foot-pounds is the work by expansion. But this leaves 497,575 foot-pounds remaining of the energy above estimated as contained in 1 lb. of steam. In theory this is supposed to be still retained in the steam on discharge; but as $\frac{365521}{772}$ = 473½ degrees of the total heat of the steam have been abstracted from it in the work done, nearly one-sixth of this loss ($\frac{1}{6}$th) having taken place before expansion commenced, it is apparent that not even a 135-fold expansion could give the full power estimated; and in any case the consumption of fuel per horse-power per hour would be nearer ¾ lb. than ¼ lb. This, of course, is assuming that none of the heat of the steam is abstracted through the material of the cylinder. The loss in this way may be such that, at very high rates of expansion, so far from realizing any gain, a positively greater weight of steam and of fuel are required for a given power than when the engine is worked at full stroke. The cylinder, it is true, may be kept hot during expansion, either by steam jacketing or by being surrounded by hot-air flues. At high rates of expansion the amount of external heat necessary to prevent liquefaction would be such as to burn the lubricating material out of the cylinder. The specific heat of steam being much less than that of water, a given pressure in expanding would be had by heating the steam with a much smaller absolute quantity of heat than would be required to effect the same pressure if the heat were applied directly to the evaporation of water. The whole theory of the expansive action of steam is based upon our knowledge of the laws of gaseous bodies; and it takes no account of the mutual attraction between the particles of steam—an attraction which, upon the loss of heat, restores them to their original condition of liquid water, thus at once lessening the pressure by enlarging the volume of the remaining steam. The confession must be made, that the knowledge embodied in the mechanical theory of heat has not as yet led to any improvement in the principles of the steam engine, beyond what was understood by a number of engineers from thirty to ninety years ago. The mechanical theory of heat, however, or rather the experiments made to establish it, have shown that when a gas expands without the exertion of work, it does not lose in temperature; and if steam were thus expanded, it would be superheated in the very act of expansion, so that although having a normal temperature of 298 degrees at a pressure of 50 lbs. above the atmosphere, it would still be at a temperature of 298 degrees, if expanded freely into a vacuous space, down to a very low pressure.

The laws governing the motion of gaseous bodies, and especially of steam, have a useful application in connection with the steam engine. The theory upon which is calculated the velocity with which an elastic fluid flows into a vacuum, or into a fluid of lower pressure, may be thus explained. It is a law of nature that water, and indeed all fluids, will stand at the same level in two vessels connected at the bottom. So, too, if a jet pipe be connected to the bottom of a vessel filled with a fluid, and the mouth of the jet be directed upward, the fluid, on escaping, will tend to rise to the level at which it stood in the vessel itself. The real ascent is never so great as this, but the difference is attributable to friction and atmospheric resistance. Leaving these out of consideration, it is easily calculable, from the established laws of falling bodies, what is the velocity with which an upward jet of water begins to rise towards the level of the source of supply. It must begin to rise with the same velocity which it would have acquired in falling freely, *in vacuo*, from that level. This velocity, expressed in feet per second, will be 8·01 times the square root of the height in feet. Thus, for a height of 100 feet, it will be $\sqrt{100} \times 8{\cdot}01 = 80{\cdot}1$ feet per second. But water has no monopoly of this law. Oil and spirit have a less specific gravity than water, and if we consider merely a given *pressure* of the fluid upon its base, a pressure of say 10 lbs. per square inch would require a higher column of oil or of alcohol than of water. Thus, if we consider the pressure only of a column of fluid upon its base, the effluent velocity will be inversely as the square root of the specific gravity of the fluid.

Now, the law applicable to non-elastic fluids holds good with gases and with steam. The readiest illustration of the flow of gases may be taken from that of air itself. The atmosphere surrounding the globe is estimated to have a height of about 50 miles, but its density constantly diminishes as we rise from the earth's surface. This diminution is in virtue of the elasticity of the air. If we conceive a vertical air-tight pipe, 50 miles high, to be first exhausted to a perfect vacuum, air admitted to the bottom of the pipe would ultimately rise to the top. But if we consider the case of a single particle of air, it would not rush in at the bottom of the pipe with a velocity which would carry it to the top, nor would the top of the pipe become filled at all, except gradually by ascension, as air of greater and greater density accumulated at the bottom, and upon which denser air that of less specific gravity would float. So far as concerns the velocity with which air would rush into a vacuum, it is to be considered as a non-elastic fluid (or, to make the comparison plainer, a liquid) of the same specific gravity. Regarded as non-elastic, and therefore of uniform density, a column of air, of the usual atmospheric pressure of 14·7 lbs. per square inch, would be almost exactly 5 miles high. The velocity acquired by a body in falling, *in vacuo*, from this height, would be $\sqrt{26{,}400}$ feet $\times 8 = 1{,}300$ feet per second, and this is the velocity with which air moves into a vacuous space: and, as the pressure of air is inversely as its density, this is the velocity, into a vacuum, of air of all pressures. For whether air, by rarefaction or compression, have a pressure of 5 lbs., 10 lbs., or 100 lbs. per square inch, the height of a column of air of either pressure, and of uniform density, will be alike 5 miles, in order to produce that pressure by its own dead weight. The air has its elastic force in addition to its weight; and if we considered only the first particle escaping into a vacuum, a greater velocity than that already calculated might be inferred. But if the first escaping particle move with a greater velocity, this can only be because it has the power of pushing itself away, and unopposed, from the particles behind it. And as action and re-action must be equal, these must be, therefore, held back in exact proportion to their own elastic force, so that, so far as its effect in generating effluent velocity is concerned, the mere elasticity of a gas is inoperative.

Steam of atmospheric pressure occupies 1,642 times the volume of the water from which it was raised, and as a cubic foot of water weighs 62·4 lbs., a cubic foot of steam of atmospheric pressure weighs $\frac{62{\cdot}4}{1642} = 0{\cdot}038$ lbs. In order to exert a pressure by its mere dead weight of 14·7 lbs. per square inch, such steam of uniform density would have to stand at a height of 55,703 feet, or more than 10½ miles. The velocity due to a fall from this height is 1,888 feet per second, which is the velocity, therefore, with which steam of atmospheric pressure enters a vacuum. If the density of steam were exactly inversely as its pressure, as is the case with air, this would be the velocity of steam of every pressure in moving into a vacuum. But while a cubic foot of steam when weighing 0·038 lb. has a pressure of 14·7 lbs. per square inch above a vacuum, a cubic foot when weighing ten times as much has not merely ten times the pressure, or 147 lbs. per square inch, but a still higher pressure—viz., 170 lbs.; or say 155 lbs. per square inch above the atmosphere. And thus, while it would require a column of steam 55,703 feet high to weigh 14·7 lbs. per square inch on its base, it would require one of 64,418 feet to weigh 170 lbs., the steam being at the respective densities of 0·038 lb. and 0·38 lb. per cubic foot. The effluent velocity, therefore, of steam of 170 lbs. total pressure will be 2,030 feet per second.

In dealing with non-condensing engines, the effluent velocity of steam into the atmosphere, or into steam of lower pressure, is to be considered. Here only the difference of pressure is effective in generating motion; and the corresponding height from which the velocity is to be calculated is that of a column of steam of the higher pressure, weighing by its own gravity as much as the difference of pressure. If, for example, steam of 1 lb. pressure above the atmosphere be discharged into the air, the height generating the velocity of escape will be that of a column of steam of 15·7 lbs. total pressure, weighing 1 lb. upon each square inch by its gravity alone. In order to conform to the table on page 111 we will calculate the effective pressures as for 1·3 lbs., 2·3 lbs., 10·3 lbs., &c., afterwards dropping the decimal. Thus, in the following table, the pressures given in pounds above the atmosphere are each 0·3 lb. less than the pressures employed in making the calculation.

Pressure above the Atmosphere.	Velocity of Escape per Second.	Pressure above the Atmosphere.	Velocity of Escape per Second.
Lbs.	Feet.	Lbs.	Feet.
1	540	50	1,736
2	698	60	1,777
3	814	70	1,810
4	905	80	1,835
5	981	90	1,857
10	1,232	100	1,875
20	1,476	110	1,889
30	1,601	120	1,900
40	1,681	130	1,909

The mode of calculating these velocities is this:—A cubic foot of steam acting with its whole weight upon an area of one square inch will form a column of steam 1,728 inches, or 144 feet high and 1 inch square. If its weight be 0·1111 lb., and its elastic force be 31·3 lbs. above the atmosphere, it follows that, in order to exert the same pressure by its weight alone, a column of steam of this pressure would require to be as many times higher than 144 feet, as the weight of 1 cubic foot is contained in 31·3 lbs. This is 281·7 times, and thus the height generating the velocity is 144 × 281·7 = 40,565 feet, and the effluent velocity from this height will be eight times its square root, or 1,619 feet per second. The same result will be obtained, and a few figures saved, by at once taking the square root of the quotient, obtained by dividing the pressure of the steam by its weight per cubic foot, and multiplying it (the root) by 96.

It will be understood that these calculations apply only to saturated steam, or steam as it rises from the water from which it was generated. Whatever the pressure per square inch, common to different conditions of steam, or to vapours, the effluent velocity will be inversely as the square root of the specific gravity of the steam or vapour. If the steam be superheated, its specific gravity for a given pressure will be diminished, and its velocity of escape into the air, or into a vacuum, will be increased. If, on the contrary, the steam carry with it any suspended moisture, its specific gravity for a given pressure will be increased, and its velocity of escape diminished.

It may be as well to point out, that as the pressure of steam rises, a given difference of pressure will generate a less and less velocity of escape into steam of the lower pressure. Thus steam of a total pressure of 16 lbs. will rush into steam or air of a pressure of 15 lbs. with a velocity of 473 feet per second; but steam of 101 lbs. will move into steam of 100 lbs. with a velocity of but 199 feet per second; the absolute weight of the steam in motion, under the same constant difference of pressure, viz., 1 lb., being nearly six times that in the former case.

It would extend the present chapter beyond its proper length to enter here upon the laws of the motion of steam through pipes. It will appear further on how completely the use of the indicator enables the engineer to proportion the steam pipes and passages of a locomotive engine, so as to cause a minimum loss of pressure in the movements of the steam towards the piston. Those who wish to devote special attention to the properties of steam will find further and valuable information in Professor Rankine's *Steam Engine and other Prime Movers.*

CHAPTER V.

DESCRIPTION OF THE LOCOMOTIVE ENGINE.

Among the readers of these pages a numerous class will derive advantage from a full and clear description of the structure and mode of action of the Locomotive Engine. A good deal of descriptive matter is contained in the first three chapters, forming the historical portion of the present work; but the author is aware that this will not meet the expectations of those who wish to take up and proceed with the study of the Locomotive from the beginning.

It will be more convenient to the reader if the present description be divided into three portions, treating successively of the boiler for generating the steam; the engine, or working machinery, for applying the steam so as to produce rotary motion; and the wheels, axles, and other parts which serve as a carriage for the whole.

Like other steam boilers, that of the Locomotive Engine is a hollow metallic vessel, steam-tight (at least, when well made), and the heat is communicated to the water through the substance of the vessel itself, as is the case in nearly all apparatus for heating liquids.* The simplest and strongest boiler for a given diameter and a given weight of metal would be a hollow sphere, which has twice the strength of the hollow cylinder of the same material, diameter, and thickness. Strictly, this comparison of the strength, respectively, of the cylinder and sphere applies only when the cylinder is so long that its strength in the direction of any given diameter is not increased by the resistance of the ends. In the case of cast-iron boilers, now in considerable use for fixed engines, advantage has been properly taken of the spherical form, —a number of small hollow cast-iron spheres, communicating freely with each other, being employed. A spherical locomotive boiler would, however, be awkward in its connection with the rest of the machine, and, as a sphere merely, its heat-transmitting surface would be insufficient to evaporate the requisite quantity of water in a given time. In adopting any other form, it is to be borne in mind that, as a hollow vessel, subjected to internal pressure, it always tends to assume the spherical form, as in the case of soap bubbles, a bladder inflated with air, or a gas balloon. Knowing this fact, the constructor of a boiler, of whatever form, other than that of a sphere, provides such ties or stays as will prevent any alteration in shape. In the case of locomotive, and indeed most steam boilers, the cylindrical form is chosen for convenience of construction and arrangement. A drum, or (although it is not exactly cylindrical) a barrel, will serve as an illustration of the "cylinder boiler." The flat ends are comparatively weak, and would yield but for tie bolts connecting them together, or diagonal or other stays securing them to the sides of the cylinder. Greater strength is obtained by making the ends hemispherical; and what are called egg-ended boilers, thus formed, are to be met with in factories. The earlier locomotive boilers, before the time of the Liverpool and Manchester Railway, had convex ends, although the convexity did not amount to that of a hemisphere.

Having the cylindrical boiler, to be partly filled with water, the next question is how to apply the heat. In some cases factory boilers, set in brickwork, are heated directly on their outsides, the fire place and grate being beneath them. The reader, if now studying the construction of the Locomotive for the first time, must not be impatient if his attention is thus apparently diverted from the subject by illustrations drawn from other forms of engines and boilers. For in order that a description like the present may be complete, it must always be written upon the supposition that whoever reads it is otherwise uninformed of the subject. And in order that he may fully comprehend it, it is necessary to give reasons as well as facts. Now, there are several reasons why a locomotive boiler is not heated, or, in the language of engineers, "fired" from the outside. In the first place, an external fire place, if made of bricks, would be quite inadmissible upon a locomotive, for it would be always shaking to pieces, and, too, unless it were of great thickness and consequent bulk, it would waste much heat by radiation. In the second place, an external fire place, if not made of bricks, so as not to burn through, would need to be surrounded with a water-space, and this should only be made of metal; in which case it should properly communicate with, and thus form a part of the boiler itself. In the third place, the external surface of a boiler, or, rather, so much of its surface as is below the water-line (and it is only to this portion that heat may be applied), would be quite insufficient for the requirements of evaporation, unless the boiler were of very great size. The arrangement of the plates, flues, or tubes through which the heat is imparted to the water in steam boilers, or, in other words, the arrangement of the "heating surface," forms almost the whole difference between different boilers intended for the same pressure. And it is as well to understand, before going further, the reason why a great extent of heating surface has to be provided in a locomotive boiler. It is found, in the working of railways, that from 100 to 200 cubic feet of water must be evaporated every hour, in order to provide the steam necessary to move an ordinary train at its usual speed. It is, moreover, a fact that, even when exposed directly to intense heat, a square foot of iron or copper plate, such as is used in boilers, cannot transmit more heat, by conduction, than will suffice for the evaporation of one cubic foot of water per hour. This is supposing that the hot air or gases playing upon the heating surface are being constantly replaced by further heated air or gases at the full temperature of the furnace. But in this case the gases are only partially cooled against the plate which separates the fire from the water. And of the whole volume of hot gases in the fire place only a portion reaches those surfaces of the boiler which are directly over or at the sides of the fire, the remainder flying, at its full temperature, towards the chimney. It would be very wasteful to allow the partially cooled gases to escape without giving up as much as possible of their remaining heat. Not only has additional heating surface to be provided, therefore, but the addition must be all the greater for this further reason:—The rate of conduction of heat through the boiler plates is in proportion only to the difference of temperature between the hot air or gas and the water being heated. Thus, if one square foot of plate will abstract say one half of the heat of the gas, and conduct this heat to the water, the remaining difference of temperature between the air or gas and the water thereby becomes so much less that the rate or rapidity of conduction is greatly diminished; and as the heat is successively lowered, larger and larger additional heating surfaces must be provided for successive equal decrements of temperature. Altogether, nearly a square yard of heating surface is requisite for the evaporation of one cubic foot of water per hour in locomotive boilers, it being understood that the hot air and gases are then discharged at a low temperature—nearly all their available heat having been taken from them by the water.

Let us now see how perhaps 1,000 or even 1,500 square feet of heating surface are provided in a boiler which is neither too bulky nor too heavy for a railway locomotive. If 1,500 square feet of heating surface were to be provided on a plain cylindrical boiler 5 feet in diameter, half-filled with water, and heated over the lower half of its external surface, it would require to be 185 feet long, and it would contain 50 tons of water. If half-inch thick, and stayed at the ends, its own weight, empty, would be 30 tons. All the heating surface of the locomotive boiler, as actually made, is, however, internal, or within the boiler itself. The several plates accompanying the present work represent nearly every form of locomotive boiler; but as it happens that mechanical drawings seldom convey a clear notion to those who are not in the habit of consulting them, and who, too, have not already seen the object itself, represented in the drawing, we shall employ another mode of illustration, only referring incidentally to our engraved sections. Take a card-paper cylinder of any size, and of a length equal to three or four times its diameter. Let it be closed at one end, and then construct from it a model locomotive boiler as follows:—From the open end of the cylinder first cut it through lengthwise for about one-third its length, and then, from the point to which this cut thus extends, cut right and left each way for one-fourth of the circumference, thus leaving the cylinder cut half across its diameter. Place the cylinder horizontally, and in such position that the two loose flaps of paper formed by the cutting just described may be turned outward from their original curve, and so as to stand upright, and parallel with each other. These two flaps shall form the sides of what we now call the outside fire box. Suppose their lower edges to stand upon a table; two pieces of card are then to be cut, the one of such form as to cover the open end of the model down to the table; the other of a different form, to complete the outside fire box between the cylindrical portion of the model and the table. The vertical and curved edges of these pieces of card will be properly gummed or glued, to make tight joints with the corresponding edges of the model. The outside fire box is now complete, and tightly jointed to the cylinder, of which a portion next one end, it will be seen, forms the semicircular top or roof of the outside fire box. Next, make a square paper box, closed at the top and open at the bottom. Its length and breadth, as measured horizontally, will be a little less than the corresponding length and breadth of the outside fire box; and its height will be such that it shall rise a little above the middle of the cylindrical portion of the model; both the model and the square paper box, which we shall now call the inside fire box, being supposed to stand on the table. The inside fire box being placed within the outside fire box, we may now refer to Plate XV., giving a longitudinal section of Mr. Ramsbottom's goods engine on the London and North Western Railway, as the clearest for the further illustration of the construction of the boiler. The outside and inside fire boxes, the former of iron and the latter of copper, are seen in section towards the right hand in the engraving. The water-space between them, and which completely surrounds the inner fire box, will be seen closed at the bottom by a square bar, also shown in section. This bar is bent and welded to the proper form, to extend around the bottom of the inside fire box, and it is rivetted and tightly caulked to both fire boxes. The water in the water-spaces is in free communication with the rest of the water in the boiler; and thus the flat sides of the respective fire boxes are exposed to the full pressure of the steam, which tends to burst the outside fire box, and to collapse the inside fire box. These flat sides, by themselves, would be unable to resist the strain upon them but as the strain upon the respective fire boxes is in opposite directions, and necessarily equal for equal areas of surface, tie bolts, or, as they are called, "stay bolts," are screwed through the plates at frequent intervals, so as to connect the two fire boxes securely together, the ends of the stay bolts being also rivetted, or spread out by hammering so as still further to increase their holding power. Thus secured, the water-spaces, according to Mr. Fairbairn's experiments, are the strongest parts of the boiler.

In dismissing our paper model, it may as well be said that the mode of cutting the paper, in its construction, as above described, is not that followed in making a boiler in iron and copper; but this mode will best serve to carry the idea of the boiler into the mind. The boiler-maker does not cut open a cylinder to make the fire box, but makes each part separately, and unites them afterwards.

The flat top of the inside fire box is of course equally weak with the sides. It could not be satisfactorily secured by stay bolts to the roof of the outside fire box, and it is stiffened, therefore, by a series of iron bars, placed on edge, and of considerable depth, and which are firmly connected to the roof or crown of the inside fire box by rivets. The roof, therefore, can only be crushed downwards by bending these bars, which are of great strength, at the same time. These bars usually extend in the direction of the length of the fire box, as shown in Plate XV.; but they may extend across it, as shown in section on the roof of the inside fire box in Plate XIX. These stay bars bear on the fire box only for a short distance at each end, iron rings or washers being interposed between them and the roof-plate at the points where the bolts or rivets which secure the bars pass through. This permits the water to circulate under the bars, and prevents the roof-plate from being burnt, as it would be if the water were excluded from the whole under-surface of the stay bars.

We have not yet filled the boiler with water, nor lighted a fire in the fire box, nor, indeed, provided any means for introducing and supporting the fuel, nor for carrying off the hot air and gases of combustion. A horizontal grate of thin and deep bars is fitted across the bottom, and, indeed, forms the bottom of the fire box, and a door is made to open into the fire box from the foot-plate. The opening for the door has to be made through the plates of both fire boxes, and in order to keep the water-space tightly closed, a ring of iron, of which the inner diameter corresponds with that of the door, is rivetted between the outside and inside plates, as shown about half-way down the hind water-space, in the right hand of Plate XV. It will be understood that the inside fire box is covered with water to the depth of about 6 inches; and that shown in the plate referred to thus has about 90 square feet of heating surface. Were it not that this surface, although it is in the best possible position for the rapid generation of steam, does not absorb more than one-third or one-half of the useful heat of the gases of combustion, a short wide outlet might be made directly from the fire box to a chimney. As it is, so large a quantity of fuel is ordinarily burnt in a given time, that the larger proportion of the heat of the gases remains to be taken up after they have left the fire box; and for the reasons already given, a much larger additional surface must be provided for the transmission of given quantities of heat, as the temperature of the gases is successively lowered, nearer and nearer to that of the water in the boiler. It is found in practice that from six to twelve times the heating surface in the fire box must be provided in order to bring down the heat of the gases to nearly that of the water: meaning by "nearly," that the excess of the temperature of the former, on leaving the boiler, shall not be more than 200 degrees or 300 degrees over that of the water. This great necessary extension of heating surface can only be obtained by means of a large number of small thin tubes, of considerable length, leading directly from the fire box, through the water, to the chimney. The reason why the necessary surface can be better provided in a number of small tubes than in a single large one, may be explained thus:—By increasing the diameter of a tube, in order to obtain additional surface, we increase the circumference in the same proportion only as the diameter; but we increase the space occupied by the tube four-fold. The tube, when laid horizontally, is not only twice as high, but it is also twice as wide, and therefore occupies four times as much room as before. Its strength to resist a bursting pressure from within, or a collapsing pressure from without, depends, within certain limits, upon there being a constant proportion between its diameter and thickness; thus, if the diameter be doubled, the thickness must be doubled also. And in doubling the circumference (by doubling the diameter), and doubling the thickness, the weight is also increased four-fold. A 12-inch tube 1 foot long has 3 square feet of surface, and it occupies nearly $\frac{3}{4}$ths of a square foot when placed upon end on a table. If it be placed within a boiler of a given diameter and cross-sectional area, it will of course also occupy $\frac{3}{4}$ths of a square foot of that area. But twelve tubes, each 1 inch in diameter, would present the same total external surface, while they would not occupy but one-twelfth of the space. And if the thickness of each tube was diminished in proportion to the diminution of diameter, the total weight of the twelve small tubes would be but one-twelfth that of the single large one. In other words, with a given total weight of tubes occupying a given total space in a boiler, twelve times as much surface may be had with 1-inch as with 1-foot tubes. This is supposing that the necessary thickness of the tubes was exactly in proportion to the diameter; but this is not the case in practice. The 12-inch tube should be $\frac{3}{8}$ths of an inch thick, and the 1-inch tube $\frac{1}{16}$th of an inch thick. Another consideration is also involved in the comparison of large and small tubes. It is held by many that there should always be a definite proportion between the area of the fire grate of the furnace and the area of the opening of whatever kind through which the hot gases escape from the fire place. The extent of this opening has been called the "calorimeter" of the boiler, although, as this word literally signifies a measure of heat, its application in this case is not appropriate, if, indeed, it be correct. That some definite area for the outlet of the hot gases is important in every given case, there is no doubt; but this area depends partly upon the distance through which the gases traverse the heating surfaces of the boiler. This can be made clear, perhaps, as follows:—Let two boilers have equal heating surfaces, and an equal area of grate, and let the hot gases have the same "run," or, in other words, let them traverse the heating surfaces of the boiler for the same total distance in each case. But while one boiler has 100 square feet of heating surface in ten tubes 12 feet long and $3\frac{3}{16}$ inches in diameter, the other shall have the same heating surface in a single flue of the same length and $31\frac{3}{4}$ inches in diameter. Now, in the two cases the heating surface and the "run" of the hot gases are the same; but the outlet for the gases is ten times larger with the single flue than with the ten tubes each of one-tenth the diameter. It can be said at once, as the result of experience with which every competent engineer is familiar, that, in evaporating a given weight of water per hour, the boiler with the ten smaller tubes would be the more economical of fuel. The fact is, the larger flue would convey so large a bulk of hot gas through it that the particles of gas towards the centre of the flue would not have time to move among themselves sufficiently to bring each particle successively in contact with the heating surfaces, where alone it would give off its heat. But were it argued that only the area of outlet opening afforded by the ten smaller tubes should be adopted, it might be said that the same area may be had with a flue nearly $10\frac{1}{8}$ inches in diameter. But to obtain the area of heating surface, viz., 100 square feet, which we have been considering, this flue would require to be 37 feet 9 inches long. Yet, from the general experience of engineers, it may be said that, were a single flue to be adopted, one 20 inches in diameter and 20 feet long would be preferred to that just considered; while, if a number of tubes were employed, fifteen of a diameter of 2⅛ inches each, and 12 feet long, would be preferred to the ten of the same length and 3$\frac{3}{16}$ inches in diameter. In all these cases the heating surface is the same, viz., 100 square feet; but so far as outlet opening is concerned, a single 20-inch tube with 314 square inches of cross-sectional area, or fifteen tubes, each 2⅛ inches in diameter, and presenting in all only 53 square inches of area in cross-section, would, with the same consumption of fuel in the fire place, give results very much the same. The 20-inch tube would have a "run" of 20 feet, and the smaller tubes a "run" of 12 feet; but no great difference would be found in the economy of fuel.

We have now explained at considerable length the reasons why small tubes are preferred to large flues for conducting the gases of combustion from the fire box through the water to the chimney in locomotive boilers. The necessary heating surface is obtained in the former case with a much more compact, and a much lighter arrangement. In locomotive engines thin brass tubes, generally not far from 2 inches in diameter, and from 10 feet to 12 feet long, are employed to conduct the hot gases from the fire box to the chimney, the number of tubes varying, according to the size of the engine, from 100 to 200, or more. The arrangement of these tubes, the uppermost row of which is covered by from 6 inches to 8 inches of water, is shown in most of the cross-sections of locomotives given in the various plates attached to the present work. The front plate of the inside fire box and the front plate of the cylindrical portion of the boiler, or that to the left hand of Plate XV., are accurately drilled, to receive the ends of the tubes, which pass through the plates, and are made steam-tight within them by means of ferrules of wrought or cast iron, which are driven into the ends of the tubes, so as to force them tightly into contact with the interior surfaces of the holes in the tube plates. Such is the tightness with which the tube-ends are thus secured in their plates, that not only is there no leakage of water, as long as the joints are kept in good order, but the tubes serve as ties to prevent the respective tube plates from being forced outwards, as they otherwise would be by the pressure of the steam. Through that portion of the boiler above the tubes a number of tie bolts extend longitudinally from the smoke-box tube plate to the back plate of the fire box, to hold these plates together against the pressure of steam, tending to force them apart.

It now remains to produce a draught through the tubes, and to conduct the waste gases escaping from them up a chimney. The tubes lead into a closed chamber, formed upon the front end of the boiler, and called the smoke box. Although the smoke box has a removable door in front, this is tightly closed when the engine is ready for working, and then there is no inlet of air to the smoke box except through the tubes, and no outlet except by the chimney. The smoke box (with its front door, to give access to the tubes when required) and the chimney will be easily recognized upon nearly all our engraved plates which give longitudinal elevations or sections of locomotives.

It need hardly be said that the natural draught in the locomotive boiler, with a chimney seldom more than 12 feet 6 inches high above the level of the fire grate, or 14 feet from the rails, would be quite insufficient for the combustion of (as is often the case) 14 cwt. of coal per hour. Nothing could, however, be more simple and effective than the means employed to excite the fire. All the steam worked through the engine escapes afterwards through a pipe with a contracted mouthpiece, which points directly up the centre of the chimney. A section of this, the blast pipe, with its mouth marked 4¼ inches as the diameter, will be seen in the smoke box in Plate XV. The steam shoots up with great velocity, expanding as it goes from the diameter of the blast orifice to that of the interior of the chimney, through which it rises, to escape finally into the air. The particles of steam in this rising jet roll and force along the neighbouring particles of air, and these, in turn, move those beyond them, until the whole of the hot air and gases in the smoke box tends towards the chimney. As the smoke box is thus partially emptied (and nothing like a complete vacuum is ever produced), the external pressure of the atmosphere forces in air, through any and every opening to the smoke box, to take the place of that already drawn out or exhausted from it. As the only inlet is through the tubes, into which the gases of combustion have free access from the fire box, and as the external air has only to pass through the fire grate, and through the burning fuel, to reach the fire box, there is a constant draught of air through the grate as long as the waste steam is shot up the blast pipe and through the chimney. It is thus that, within certain limits, the more the steam that is required, the more the steam that is produced; for all the steam used in the engine draws in the air in its final escape, to excite the fire to generate more steam. In first raising steam from cold water in the boiler, the fire is maintained only by the natural draught of air through the grate, tubes, and chimney; and it thus takes a long time—an hour or more—to raise the steam with which the engine starts on its journey.

The locomotive boiler has further to be provided with apparatus for supplying it with water, and for ascertaining that this supply is neither too much nor too little; means also for preventing the pressure of the steam from rising beyond that required for the working of the engine; besides various fittings of minor importance.

Before steam is raised, and when the boiler is empty, it is first filled with water to the height of a few inches above the fire box, by means of a hose connected with a cock placed on any convenient part of the boiler. In order to know when the water is at the right height there are two "gauge cocks" fixed in the back plate of the fire box, towards the engineman's foot-plate, the one cock a few inches above, and the other as much below the proper level of the water within the boiler. These cocks have steam-tight fittings connecting them with a glass tube, within which the water, having free access from the boiler through the lower cock, is free to rise and fall, the surface of the water in the glass being under the pressure of the steam, freely admitted from the boiler through the upper cock. The water within the gauge glass thus has the same level as that in the boiler; and the engineman has only to look at this glass to see what the height of the water is. This glass is almost the only means relied upon to show the height of the water in the boiler when the engine is working. In some cases, however, a series of three or more cocks is fixed in the back plate of the fire box, and at various heights between the extreme limits admissible for the water level. By trying these cocks successively, the engineman can judge, according as steam or water comes out, at what height the water stands. When working, the boiler is supplied with water either by one or a pair of force pumps, or by an instrument invented a few years ago, and known as the Injector, a description of which has been already given on page 100. The Injector is now rapidly taking the place of force pumps for feeding water to steam boilers. Where force pumps are still used, they are worked from some convenient moving part of the engine itself.

Two openings are made in the upper part of the boiler, and these are covered by discs or valves, which are held down each by a lever, one end of which is secured directly to the boiler, while the other end, generally much the longer of the two, is held down by a stout spring balance, screwed to such a degree that the end of the lever, and thus the disc or valve, can only rise when the pressure of the steam in the boiler becomes greater than the balance or spring scale can resist. These valves are called safety valves, because, by rising when the pressure of the steam exceeds the intended limit, they allow it to escape into the air, thus preventing any excessive accumulation of pressure whereby the boiler would be perhaps endangered. These valves are distinctly shown in several of our plates, particularly so in Plate IX., one over the fire box, and the other over the middle of the boiler, at the top of the dome. But the safety valves cannot show how much the pressure of the steam may be below the proper limit; nor, indeed, how much above it, otherwise than as can be judged by their furious discharge of steam. Enginemen once held it to be a good sign that they were attending properly to their business when their safety valves were roaring for the whole time the engine was at work; and indeed, Pambour, thirty years ago, estimated that one-fourth of all the steam generated in the Liverpool and Manchester engines was lost by blowing off at the safety valves. Within the last ten or twelve years, however, locomotives have been fitted with instruments by which the pressure of steam is shown on a dial. These are known as steam gauges, or pressure gauges; and as they are appendages only to the engine, and are never made by the engine-maker, they need not be further described at present. They are placed in such a position on the engine that the driver can see at a glance what is his pressure of steam; and to save waste at the safety valves, the pressure is not intentionally permitted to rise quite to the limit to which these valves are screwed down.

To prevent cinders and live coals from falling through the fire grate upon the line, and partly for another reason, which will be presently mentioned, a plate-iron tray is fixed beneath the fire box, and a few inches off the rails. This is the ash pan, clearly seen in Plate XV., as well as in several other of our engraved illustrations. It is often important, when the engine is standing, to prevent any access of air to the fire box; and hence the ash pan is made to fit tightly to the fire box on all but the front side. This side is opened or closed at pleasure by a hinged plate, called the damper, and which is adjusted by a rod worked from the foot-plate. When the engine is running rapidly, with the damper open, a slight advantage is also gained by the rush of air into the ash pan. At 60 miles an hour, or 88 feet per second, the pressure of the air against the moving surface would be over one ounce per square inch, or 9 lbs. per square foot. For countries where much snow falls, it is necessary to have a damper also at the hind end of the ash pan, as otherwise it would soon become choked with snow when there was more than a few inches in depth of this upon the ground. In going forward, the front damper is then closed and the hind one opened.

The construction of the boiler will by this time have been made sufficiently clear to the mind of the reader; and he will also have become acquainted with the reasons why the boiler is made as it is. We have described it only so far as is necessary to make it a complete steam generator, and without reference to the parts which connect it with the other portions of the engine. It will be sufficiently understood that the boiler is made of plates of iron and copper—the latter for the inside fire box—and that these plates, ⅜-inch thick and upwards, are bent or cut, as required, to the necessary forms, and secured together by rivets tightly caulked. The boiler is furthermore covered with wood or felt, or both, to prevent the loss of heat by radiation; and a casing of thin sheet iron is placed outside of all, to give a good finish, and to protect the wooden "lagging" or "cleading" from injury by the weather, sparks, or otherwise.

The steam, now supposed to have been generated, is next to be applied to produce motion in the engine. Although the term "engine" is commonly applied to the boiler and working machinery collectively—at least in the case of railway engines—it will be understood, in the following portion of the present description, that the term is applied more particularly to the working machinery only. The first thing is to take the steam from the boiler; and locomotive boilers, evaporating so large a quantity of water within so small a space, are apt to mingle a good deal of spray with their steam. It is important, for two reasons, to take only dry steam from the boiler. Practically, whatever water is carried along with the steam, and heated, of necessity, to the same temperature as the steam, carries off so much heat to no useful purpose with it. Next, if the quantity of water be considerable, its presence in the working machinery may lead to serious mishaps, as will presently be understood. There are other reasons, too, which will be given further on, why the steam should be dry on leaving the boiler. One condition which must be fulfilled in order to obtain dry steam is, that it shall be taken off at a sufficient distance above the water level in the boiler. Experience only can show what this distance is. Somewhere about 3 feet is found to answer well, and hence, as the water, in locomotive boilers is maintained at a level generally within 1 foot of the top of the cylindrical barrel, a sort of cupola, or, as it is commonly called, a dome, has to be erected for the purpose. Referring again to Plate XV., this dome will be seen midway between the fire box and chimney, on the top of the boiler. The dome is thus a part of the boiler itself, and it is in shape like a hat, rivetted down by its brim around an opening cut in the cylindrical barrel. It is from near the top of this dome that the steam is taken from the boiler, except where, as in Messrs. Hawthorn's arrangement, described on page 69, and illustrated in Plates III. and IV., it is taken off from a long pipe perforated with a great number of small openings. This pipe, called the steam pipe, has a stop valve either at its mouth in the dome, as shown in Plate XV., or where it enters the smoke box. It is a pipe merely intended to take the steam from the boiler to the engine, and having means for turning on and shutting off, as well as regulating, the supply of steam when required. It was once the practice to place the safety valves on the dome. So long as they do not blow off steam, it matters little where they are placed, so that they are above the water-level. But in blowing off strongly, the steam, rising to the safety valves, carries water with it; and if they are just over the mouth of the pipe which supplies the engine, some of this water will be carried along with the steam. So the safety valves are now seldom placed upon the dome, but, as in Plate XV., over the fire box.

Having arrived thus far, it will be better, instead of following the steam inch by inch from the boiler, to examine at once its mode of action in the engine. Although steam does not act with the explosive force of gunpowder, it is probable that its action will be best understood by a reference to ordnance. A cannon is a strong hollow cylinder, closed at one end, and from which a heavy shot of metal is projected at the other by the expansive force of fired gunpowder. And a steam engine can be best compared to a cannon. If, instead of fired gunpowder, we introduce high-pressure steam between the closed end, or breech of the cannon, and the projectile, we shall drive the latter out at the muzzle. Jacob Perkins projected balls in this manner from his high-pressure steam gun in 1829. But in a steam engine the steam is conducted from the boiler into a cannon which has a breech at both ends, and within which the ball is shot alternately from one end to the other. This, it might be said, would be impracticable; for, as a gun, it would burst. But if the shot be connected, by means of a strong rod, with a crank outside the gun, it will be seen that, as long as the rod and crank do not break, the shot cannot strike the breech at either end. We will now drop the comparison with the cannon, which we will call the cylinder, and the cannon ball shall be known as the piston. The breech at either end is a cylinder cover, and the rod, extending through one of these covers towards a crank, is a piston rod. The cylinder is smoothly and accurately bored in its interior, and the piston, instead of being a solid block of metal merely, has elastic metallic rings around its circumference. These rings are originally made a little larger than the bore of the cylinder, and are then cut open at one point in their circumference, and so much of their circumference is removed in this cutting that they may be compressed sufficiently to enter the cylinder, within which they tend always to press equally and closely against the whole circumference of the bore, thus insuring steam-tightness even after the cylinder and piston have become slightly worn.

The working machinery of the engine may be better explained from Plates III. and IV. than perhaps from any others in the present work. Here only the middle or larger pair of wheels, known as the driving wheels, are to be turned by the steam machinery, the smaller leading and trailing wheels rolling in the manner of ordinary carriage wheels. In Plate III. the longitudinal section, occupying the middle of the plate, is incorrectly lettered "longitudinal elevation." In this section the steam pipe will be seen close to the upper part of the boiler, the pipe being drilled through, on its upper side, with a great number of small holes—too small to be shown in the engraving,—and through which holes the steam enters the pipe on its way to the cylinders. The stop valve is in this case in a cast-iron box in the smoke box, and is merely a flat plate of metal, made to slide at will upon a corresponding flat surface formed on the bottom of the valve box, which is always open to, and may indeed be regarded as an extension of the steam pipe. By moving the stop valve, in this case by hand, to the right or left, an opening is uncovered, which permits the steam to pass into a pipe plainly shown in the "section through cylinders," being the right-hand upper figure of Plate IV. In this figure the pipe taking the steam to the cylinders is shown as bending from the upper side of the smoke box, down and around the left-hand side of the smoke box to the slide-valve box, a little below the lower tubes. There are two cylinders to almost all locomotives, and in this case the slide-valve box, which is always filled with steam when the engine is working, is placed between them. In the plan, or lowermost figure, of Plate III. both cylinders are shown towards the left-hand end of the engraving,—one in section, or cut lengthwise through the middle, as a barrel might be supposed to be cut open; while the other cylinder is shown by a few lines which, although they give perhaps no idea of its complete form and purpose, yet truly represent its exterior as viewed from above. The cylinder in section is marked 16 inches in diameter and 22 inches stroke; and it may be observed here that the engine from which it is drawn was that exhibited by Messrs. Beyer, Peacock, and Co., in the International Exhibition of 1862, and built for the South-Eastern Railway of Portugal—a line of 5 feet 6 inches, or "medium" gauge. In the cylinder shown in section will be seen the piston at the right-hand end, the piston being fixed on its piston rod, and fastened by a large six-sided nut, shown in the section, and also in the "section through cylinders," Plate IV.

It is now time to see how steam is admitted, first to one side of the piston and then to the other, and how, when the piston has been driven by the steam to either end of the cylinder, the steam which has thus done its work is permitted to escape at the same time that fresh steam is admitted upon the opposite side of the piston. For it is by an alternate and repeated admission of steam to the two ends of the cylinder, and the instantaneous release of the steam at the moment when it has forced the piston to the end of its stroke, in either direction, that the to-and-fro motion of the piston is maintained; and it will be presently shown how this motion is imparted to the axle of the driving wheels, so as to turn them either forward or backward, according to the direction in which it is required to work the engine. In the cylinder shown in section, in the "plan," Plate III., a bent passage or port will be seen leading from each end (on the lower side in the drawing), and terminating on a straight line, which in this case is marked as 11⅛ inches from the axis or centre line of the cylinder. Where these passages or ports open upon the straight line just named, a third passage or cavity will be seen between them. The last-named passage has no direct communication with the interior of the cylinder. In the longitudinal section, just above the plan, the three passages will be seen, partly in front and partly above the circle representing the leading wheel. They open out upon a flat face formed on the side of the cylinder, and are about 13 inches each in height, in this instance; while the two passages or ports from the ends of the cylinder are perhaps 1¼ inches wide, the middle passage being about 3 inches wide. These passages open into a box between the cylinders, which, as explained above, is always filled with steam when the engine is working. It is easy to understand, however, that by placing a flat square plate of metal against the flat face of the cylinder, and by taking care that this plate is large enough to cover the openings of the three passages, steam could be altogether excluded from them, and thus excluded from the cylinder itself, it being understood that the flat plate in question fits steam-tight upon the flat face of the cylinder through which the three passages open. Once in contact with the flat face of the cylinder, this plate would be pressed closely against it by the pressure of the steam. But if the plate be moved so far in either direction, from or to the forward end of the cylinder, as to uncover one or the other of the passages leading into the cylinder, steam would be instantly admitted; and if the plate were worked to and fro while in contact with the flat external face of the cylinder, and for the proper distance in each direction, steam would be admitted alternately to the opposite ends of the cylinder, and thus to opposite sides of the piston.

But in order to keep the piston continuously in motion backwards and forwards, it is necessary to let off the steam, from one end of the cylinder, at or before the instant when fresh steam is admitted at the other. That is, the steam which has done its work must be allowed to escape from say the front side of the piston, before steam is admitted to the back side; and after the piston has then made a stroke from the back to the front, the steam must be let off from the back side before the admission of fresh steam to the front side. In some forms of steam engines separate steam passages and separate valves are provided for this purpose; but in nearly all high-pressure engines, including locomotive engines, the valve which controls the admission of steam to the cylinders is also made so as to permit its escape when it has done its work. For this purpose, instead of being merely a flat plate, the valve has a cavity or recess formed in its working face, or that next to the cylinder. This cavity does not extend to either of the edges of the valve, as, if it did, fresh steam from the boiler would enter the cavity, which is intended only for the spent steam. The valve is shown in section, by dark shading, against the face of the cylinder, in the "plan" or bottom figure of Plate III., and the valves to both cylinders are shown between them, and also in section, in the "section through cylinders," Plate IV. In the section in Plate III. the right-hand edge of the valve is where the right-hand port is slightly uncovered, and steam may be supposed to be entering upon the right-hand face of the piston, which is close to the cylinder-cover. The left-hand port is wholly covered by the valve, and, in the position shown, is in full communication with the cavity in the valve; and, furthermore, through this cavity, with the middle or widest passage or port. Thus, whatever steam may be in the front or left-hand end of the cylinder, is in free communication with this port, and, as will now be shown, with the atmosphere. For in the "section through cylinders," in Plate IV., this port will be seen extending partly around each cylinder, bending thence upward, and joining a pipe which rises towards the chimney, the upper open end of this pipe—the blast pipe already mentioned in describing the boiler—being a little above the upper row of tubes in the boiler. Thus, whenever the slide valve has placed either steam port of the cylinder in communication with the middle or exhaust port, the steam in the corresponding end of the cylinder has an unobstructed road up the chimney. And although the explanation has so far been applied to but one cylinder, it will be understood as applying to both. It will be seen that it is necessary that the slide valve be at least so long in the direction of the length of the cylinder (or in the direction shown in Plate III.) that it shall never leave both steam ports uncovered, else the piston would be under the full pressure of the steam on both sides, and even if the engine were not stopped, a loss of steam would result. Nor must the cavity in the valve be so long, in the direction of the length of the cylinder, as to leave both steam ports at any time uncovered to it, and thus to the exhaust port, as then the steam which ought to be at work, driving the piston, would be escaping from one or the other end of the cylinder before its proper time. It will be seen in Plate III. that the cavity in the valve, as measured in the direction of the length of the cylinder, is of such length that it can never communicate with both steam ports at once; and indeed, when it is at the point of opening one of these ports to the exhaust port, the other port will be more than covered. It will also be seen—and the more clearly if the parts be measured with the dividers or otherwise—that the valve is longer than the distance between the outer edges of the respective steam ports. It will be presently shown how the valve is moved to and fro in the direction of the length of the cylinder, or from right to left and left to right, in the plan, Plate III.; and it will be easily understood that when, in this to-and-fro motion, the valve has just covered either steam port, so as to exclude the steam from that end of the cylinder, a certain interval must elapse before the opposite end or edge of the valve can uncover the other steam port, to admit steam to the opposite end of the cylinder. The length or duration of this interval will depend upon the rate or speed at which the valve is moving, and upon the distance by which the total length of the valve exceeds that between the two outer edges of the respective steam ports. During the time while the valve is moving in either direction, through this excess of length, steam will be excluded from both ends of the cylinder, and that already in either end of the cylinder will be expanding as the piston moves before it, and thus exerting an additional amount of power beyond what would have been exerted by the same steam, had it been constantly followed up by fresh steam from the boiler. When the valve is placed midway, in its to-and-fro motion, the distance by which it overlaps each steam port is called the lap, and upon the proportion which the extent of this lap bears to the whole distance through which the valve moves will depend the interval in each stroke of the piston, during which no steam is admitted at either end, and during which, therefore, the steam already in the cylinder will be acting by its own expansion in driving the piston. It will be understood that the movement of the valve must be so timed or regulated with that of the piston that steam shall be admitted when the piston is at, or nearly at, the beginning of a stroke, and that the steam thus admitted shall be released when, or a little before, the piston has completed that stroke.

A better idea of the movements of the steam into and from the cylinder will be obtained from a flat paper model, cut to the section of the cylinder, as shown in the engravings, with a separate and movable piston and valve which may be moved by the fingers to and fro.

It will be by this time fully understood how the piston is driven to and fro in the cylinder by the steam, although we have not yet described how the necessary movements of the valve are effected. The piston being fixed tightly upon the end of the piston rod, the latter, of course, moves with it, and by a rod jointed to the end of the piston rod, and called the connecting rod, the to-and-fro motion of the piston rod is converted into the circular motion of a crank. In the present case the crank, or rather the cranks—for there are two, as there are two cylinders and pistons—are formed in the axle of the driving wheels, the cranks being in form like those made in the driving shaft of a foot lathe, or of a power loom. These cranks are visible in the plan, Plate III., but they are placed at right angles to each other, one crank (in the engraving the uppermost one) being horizontal, while the other one is upright, although it will be understood that the terms "horizontal" and "upright" refer only to the positions shown on the plan, as when the engine is working, the cranks are constantly turning around in a circle. The crank shown at full length in Plate III. (and marked 11 inches from the centre of the crank wrist to the centre of the axle, the wrist being marked 7¼ inches in diameter and 4½ inches wide) is in that position where, when the piston begins to move towards the left-hand end of the cylinder, the connecting rod (shown in section, and marked 6 feet from centre to centre) will not tend to turn the crank around in either direction, but will simply force the axle against its supports in the frame. No motion could result in this case until the crank was turned partly around, so that its own direction, as referred to the axle, should not exactly coincide with the direction of the connecting rod. In the longitudinal section, Plate III., the connecting rod and crank can be seen lying in the same straight line; and, therefore, where, but for some provision, no pressure of steam on the piston could put the crank into revolution. It is because the engine might often stop in this position that two cranks at right angles to each other are provided, so that, when one is "on the centre," as the position just described is commonly called, the other shall be in a position where the connecting rod is acting upon it with its utmost effect. It will be understood, therefore, that when one piston is at the end of its stroke, the other is at the middle, or very near the middle. As the cranks are exactly at right angles to each other, and as the position of the pistons at any moment, with reference to the length of the stroke, determines the position of the connecting rods, which in a certain sense govern the movements of the pistons themselves, we may see why the piston in either cylinder is not exactly at the middle of its stroke when its corresponding crank stands half way between the points which it occupies when the piston is at the respective ends of its stroke. And as the piston always, and necessarily, has a fixed distance from the joint by which the piston rod and connecting rod are connected together, we may refer the explanation to this joint, called the cross-head pin, as this joint will equally be at full stroke or mid stroke with the piston itself. Whatever the length of the connecting rod—a length which, irrespective of looseness in the joints, must of course be the same at all parts of the stroke of the piston—the cross-head pin must be always at a distance equal to this length from the wrist of the crank. By marking the positions of the cross-head pin, at the ends respectively of its strokes, and by then applying the length of the connecting rod from the crank, when at mid throw, it will be seen that the corresponding position of the cross-head pin is not exactly at mid stroke, but a little more in the direction of the crank. This is particularly mentioned here, not from any special importance in the fact itself, but because, while it has its bearing on the arrangement of the valve gear, it is a point which beginners in the study of the locomotive are often a long time in learning, when the fact is left to their own discovery. When the piston is exactly at mid stroke, the crank, whether standing up or hanging down, is inclined a little towards the cylinder, and this inclination will be greater as the connecting rod is shorter in proportion to the length of the crank.

As, in the progress of the engine, the connecting rods are alternately moving above and below the axis of the piston rod, or the straight line in which the piston rod moves, the outer end of this rod must be guided in its proper course. It is fixed to a piece of metal called the cross-head, and which has also fastened within it the pin upon which the end of the connecting rod is jointed. The cross-head of each piston rod is clearly seen in the plan, Plate III.; that for one piston rod being shown in section, while that for the other is shown externally. To the cross-head pin are fixed a pair of guide blocks, which slide to and fro between flat guide bars, or, as they are often called, "motion bars," although they are fixed, and have no motion. In the engine shown in Plates III. and

IV. there are four of these bars to each piston rod, there being two on each side of the rod; their positions being shown by dots just within the cylinders in the "section through cylinders," Plate IV. When the engine is going forward, the pressure of the cross-head is always upward upon the upper bars, except only when the piston is at the ends of its strokes. When the engine is working backward, the cross-head, with the exception only just named, is always pressing downward upon the lower bars. The reason why the pressure is thus constant in one direction, instead of being alternately upwards and downwards, as the connecting rod moves above and below the axis of the piston rod, will be seen on a little examination. If the crank be standing upright, above the axle, it will, when the engine is working in forward gear, turn towards the cylinder; and hence the steam will be pressing the piston towards the forward end of the engine. In other words, the steam will be pulling the crank over; and as the resistance opposed is at the wrist of the crank itself, the tendency is to pull the piston rod and connecting rod into one straight line, and thus to force the outer end of the piston rod, and thus the cross-head *upward*. When the crank, however, has gone below the prolongation of the axis of the piston rod, or when, for illustration, the crank is hanging vertically below the axle, the steam is on the front side of the piston, and is pushing the crank. But as the resistance opposed at the crank would, if it were possible, keep it in its position, the first tendency of the pressure of the steam on the piston is to push the cross-head end of the connecting rod upward in an arc, described from the crank itself, with a radius or "sweep" equal to the length of the connecting rod: so, whether the crank is up or down, as referred to the prolongation of the centre line of the engine, the strain upon the guide bars is always upward when the engine is running forward, excepting only when the piston is at the ends of its strokes, and when, consequently, no pressure, other than the dead weight of the cross-head and its attached parts can act upon the bars. By reasoning in the same manner, the pressure upon the guide bars will be found to be always downward when the engine is backing.

It has not yet been shown how the piston rod works steam-tight through the back cylinder cover. The opening through the cover is of the size of the piston rod; but this opening is enlarged within a short pipe just outside of the cover, and which is, indeed, cast or formed with it. This pipe is called the stuffing box, and being first filled with hemp soaked in melted tallow, or with some of the patented packings for the same purpose, it is closed by a short tube, or "gland," which is screwed forcibly down against the packing, so as to compress this tightly around the piston rod. A section of the stuffing box and gland will be seen in the back cylinder cover of the cylinder shown in section in the "plan," Plate III. Wherever any sliding rod has to work from the outside of the engine into any space filled with steam, or with water under steam pressure, similar stuffing boxes are employed. There are at least nine of these, in all, in the engine shown in Plate III.

Before describing how the slide valves are moved, it will be as well first to distinctly understand *how*, by causing the driving axle to revolve, the engine advances upon the rails. This, it may be said, is a result of course; but there is more in it than may at first appear. The rail is the fulcrum upon which all the power of a locomotive is exerted; and all its motions, with reference to the train, or, in other words, to space, must be referred to this fulcrum. Many an engineman has perceived this from his own observation, and has argued from it that an engine must be able to start a heavier load when the cranks are up or above the axle, than when they are below it. Now, although this conclusion is wholly wrong, it is supported by a plausible induction; and so plausible is this, that some enginemen will have it that they *know* that they can get away better with a heavy load by starting with both cranks above the axle. They have, they say, a longer leverage, the rail being the fulcrum. The reason for this notion, although it is wholly fallacious, is worth examining.

Let us suppose an engine with 2-feet stroke of piston and 6-feet driving wheels. The crank, from the centre of the axle to the centre of the wrist grasped by the connecting rod, is 12 inches long, and the radius, or half-diameter of the wheel, is 3 feet. Let us, for the sake of simplicity, suppose but a single piston, and that the total pressure of the steam upon it is 10,000 lbs. When the crank stands upright over the axle, and is being pulled over by the piston in the direction to take the engine forward, we then have 10,000 lbs. exerted at the end of a lever 4 feet long from the rail, the resistance to be overcome at the axle being at the end of a lever 3 feet long, and therefore 13,333 lbs. But this is not the force with which the engine is moved forward; for opposed to it is a force of 10,000 lbs. exerted against the back cylinder cover, and transmitted through the framing of the engine to the driving axle. So only 3,333 lbs. of effective pressure is exerted to take the engine forward. Let the crank next be down, or hang vertically beneath the axle. If it were pushed from the cylinder by a force applied externally to the engine, just as we might, when standing upon the ground, push the wheel of a carriage with the hand, the engine would be rolled backward. But instead of the 10,000 lbs. pressure upon the piston being effective in this manner, it is exerted at the end of a lever 2 feet long from its fulcrum on the rail, while it is opposed by a force of 10,000 lbs., exerted in the opposite or forward direction, through the front cylinder cover, framing, and axle boxes, and acting at the end of a lever 3 feet long. The backward pressure exerted by the piston at 2 feet from the rail, produces a backward pressure of 6,667 lbs. at the axle, 3 feet from the rail; but this is overcome, with an excess of 3,333 lbs., by the pressure exerted upon the front cylinder cover and through the framing. So the forward pressure, effective for moving the engine as a whole, is 3,333 lbs. in either case—that is, whether the crank stand above or hang below the axle. The forces exerted through the cranks upon the driving axle may be perhaps better understood if we imagine the axle not to be closely embraced by its supports, but to have a considerable "slack" in its brasses, so as to permit of the direct movement of the axle, to a certain extent, independently of the pressure transmitted from the cylinder covers. Thus, with the crank down, and with an inch clear space between the surface of the axle bearing and its support towards the hind end of the engine, so that the axle could "come and go" 1 inch without taking the engine with it, we should in the case already considered absolutely roll back the driving wheel, with a force at the axle of 6,667 lbs., and we should pull forward all that part of the engine over the axle with a force of 10,000 lbs. It would be only when the "slack" between the axle and its support was taken up that the wheel would cease to roll back, and that the effective force, in a forward direction, would be 3,333 lbs. It must not be hastily supposed, from what has been said, that the total pressure of the axle against its support is, or can possibly be, greater or less than the pressure of the support against the axle. For in the above case we have computed a backward pressure of 6,667 lbs. of the axle against its support, and a forward pressure of apparently 10,000 lbs. through the same support against its axle. But, as the force exerted and the total resistance overcome must in all cases be equal to each other, an engine advancing with a force of 3,333 lbs. must be overcoming a resistance to that amount. This resistance may be represented by a weight of 3,333 lbs., hung over a pulley external to the engine, and attached by a chain to the back of the engine framing. So that, in the case when the crank is down, 3,333 lbs. of the 10,000 lbs. pressure upon the front cylinder cover will be directly expended upon the weight being lifted (or the train being drawn), and 6,667 lbs. only will be expended upon the opposite and equal resistance of the axle, which is being pushed backward by the steam acting through the piston rod and connecting rod. When the crank is above the axle, in the case supposed, the axle will be drawn forward with a force of 13,333 lbs., while the steam pressure upon the back cylinder cover, pushing the engine bodily backwards over the axle, is 10,000 lbs., in addition to the weight of 3,333 lbs. hung over the pulley, and which weight we have taken in this case to represent the resistance of the train. So in the case supposed we have a pressure in opposite directions between the axle and its hind support of 6,667 lbs. when the crank is down, and of 13,333 between the axle and its front support, or the front of the axle box, when the crank is up, the effective pressure applied to the train being 3,333 lbs. in both cases, as already shown. In this case, therefore, the wear on the front side of the axle box and on one side of the axle will be twice as much as on the back of the axle box and on the opposite side of the axle. This great excess of pressure and consequent wear on the front of the axle box, in running forward, has led many to suppose, without due consideration, that the axle is always pressing forward in its box when the engine is going forward. There is therefore a somewhat plausible foundation for the notion occasionally entertained among enginemen, that they can start a train with more ease when the cranks are up than when they are down. The pressure exerted by the steam directly against the axle is then greater, as we have seen; but the pressure on the axle, when the crank is down, is not only less, but it is in an exactly opposite direction; and were the wheels unconnected with the engine otherwise than by the connecting rods, they would be rolled backwards instead of being pulled forward. But the pull of the engine at the draw iron is, as has been seen, the same when the crank is up as when it is down. Although we speak of the rail as the fulcrum upon which the forces exerted through the wheel are made effective, it will be understood that it is by the weight with which the wheels press upon the rails, and by their consequent friction, that a real fulcrum is obtained. As long as they do not slip, any motion in the wheels must carry the engine bodily forward, and it is necessary, therefore, that the friction of the wheels upon the rails be always at least equal to the greatest tractive force exerted by the engine. In the case which we have been considering, that of a single piston with 10,000 lbs. steam pressure, 2-feet stroke and 6-feet wheels, the greatest tractive effort is 3,333 lbs., and the friction of the wheels upon the rails must be at least equal to this. The constant tractive force is not, however, 3,333 lbs., as at the ends of its strokes the piston exerts no useful force whatever—the effect diminishing from mid stroke, where it is the greatest. The mean tractive force will, in this case, be 2,122 lbs.

The mode in which the slide valves are maintained in their proper motion, to admit the steam to, and release it from the cylinders, remains to be explained. This motion could be given by short cranks on the revolving axle of the driving wheels, each valve being connected to such a crank by suitable rods. By short cranks are meant cranks giving only the small amount of motion, say 5 inches, required to work the valves. It would be exceedingly difficult, as well as costly, however, to form such cranks in an axle of several inches diameter. What is called the eccentric forms a cheap and convenient substitute for these cranks, and in its action it may indeed be considered as a crank. Although the usual spelling is eccentric, it is literally *ex centric*, or, out of centre. The eccentric, of which there are four in most locomotives, is a circular plate of metal, fixed fast around the axle in such manner that the centre of the plate shall be at some distance to one side of the centre of the axle. In the longitudinal section in Plate III. the axle of the driving wheel is shown in section, and around it is seen one of the eccentrics, another appearing partly behind. The lines connecting the centre of the axle with the centre of the eccentric are shown in the section of the axle. The full part of one eccentric is shown over, and that of the other eccentric beneath the axle; but as the eccentrics turn with the axle, these particular positions of over and under do not apply in all other positions of the axle. It is merely necessary that they maintain always the same position in relation to the cranks. As already said, the eccentrics are, to all intents and purposes, cranks of a length represented by the short inclined lines within and to the left of the centre of the axle shown in section in the longitudinal section, Plate III. They are cranks of which the wrist, or part grasped by the connecting rod, is so large that it encircles even the shaft or axle in which the crank is formed. In the present case the cranks to which the pressure of the steam is applied through the connecting rods are 11 inches long between centres, and the wrist is 7¼ inches in diameter. If the eccentrics be considered as cranks, it will be found by the scale that they are 2¾ inches long between centres, and that they are 15 inches in diameter. The eccentrics revolve within hoops formed on the ends of the eccentric rods, (the latter marked 3 feet 9¼ inches in length in the longitudinal section, Plate III.) The eccentric rods are moved by the eccentrics to and fro; and in respect of the nature of this motion, it is identical with that of the connecting rod, except that the eccentric rods are moved by the eccentrics, whereas the connecting rods move the cranks. The ends of the eccentric rods are connected, by means to be presently described, to the stems or spindles on the slide valves, and thus move them to and fro. Thus when the full part of one eccentric is turned towards the front of the engine, the valve worked by that eccentric will be at the end of its course towards the front; and when the full part of the eccentric is exactly above or exactly below the axle, the valve will be in the middle of its throw. It has been already shown that the valve must move with respect to the movements of the piston—that is, that it must be ready to admit the steam to one end of the cylinder at the moment when the piston has reached that end, and is ready to receive the steam. The valve should of course move so as equally to uncover the ports leading to the opposite ends of the cylinders—that is, if the port leading to the front end is opened to its full width, so, on the next stroke, should be the port leading to the back end. Or if the front port is opened only to the extent of say three-fourths of an inch, so should the opposite port be opened to the same extent. This is necessary in order that the time during which the respective ports are open may be the same in each double stroke. Now, if the valve were exactly of the length represented by the distance between the outer or extreme edges of the two steam ports, it would, when placed in the middle of its throw, exactly cover both ports, and it would be equally ready, on the least motion in the proper direction, to open either. Thus, if moved forward it would open the hind port, and if moved backward it would open the front port. Supposing, further, that it was not necessary to open either steam port *before* the piston was at the corresponding end of its stroke, and ready to receive steam through that port, then the valve, if of the length above supposed, would always be in the middle of its throw when the piston was at either end of its stroke. And in order that the valve be in the middle of its throw, the eccentric which moves it would require also to be midway in its motion to and fro. And as in the engine shown in Plate III. the crank is always horizontal when the piston is at either end of its stroke, the full part of the eccentric working the valve would then be exactly above or exactly below the axle. Thus for a valve of the length above supposed, and which should not open the steam ports *before* the piston was at the end of its stroke, the eccentric for working the valve of either cylinder should be fixed around the axle exactly at right angles to the crank working in connection with that cylinder. But the full parts of two eccentrics might be fixed so as to project from diametrically opposite sides of the axle, and yet both might be at right angles to the crank. One of these eccentrics would so work the valve as to take the engine ahead; the other would, if it alone were connected with the valve, work it so as to move the engine backward. It is necessary only to examine the drawing, to see which eccentric would work in going forward, and which is going backward. In the longitudinal section, Plate III., the end of one of the cranks is seen to the right of the axle. To work the engine ahead, this crank must turn upwards, and over to the left of the axle. The eccentrics must, being fixed to the axle, turn also. Considered as a crank, it will thus be seen that the eccentric of which the whole is shown in the section, and of which the full part is uppermost, would, in turning, push the valve to the left, so as more and more to open the right-hand steam port to admit steam to maintain the proper motion of the crank in going forward. This, then, is the eccentric for forward gear. The other eccentric, of which but a part is seen, and of which the full part hangs below the axle, would only move the valve to the left, to admit steam to the right-hand end of the cylinder, when the crank turned downwards, or in the direction for going backwards. This eccentric then is the one for back gear.

It will be seen, however, that the two eccentrics shown in the longitudinal section in Plate III. are not fixed diametrically opposite each other on the axle. The lines connecting their own centres with the centre of the axle, and which may be called the radii of the eccentrics, are seen to both incline to one side of the line in which they would be at right angles to the crank. Thus neither of the eccentrics is at the middle of its throw when the piston is at the ends of its strokes. The reason is that the valve is longer than the distance between the outer or extreme edges of the steam ports, so that when placed at mid throw it would cover both ports; whereas it is necessary that steam should be entering, or ready to enter, one of the ports when the piston is ready to change its own motion from one end of the cylinder to the other. This can only be done by pushing the valve over towards one side, and as the valve is controlled wholly by the eccentric, this must be turned around and fixed in the proper corresponding position on the axle. If the valve be connected directly to the eccentric, and receive its whole motion, then the centre of the eccentric, at the extremity of the radius shown in the section of the driving axle, must be placed as far to one side of its position in mid-throw, as it is desired that the valve shall be to one side of its own mid-throw when the piston is at the end of its stroke. It will be easy, on inspecting the drawings, to say on *which* side of the position in mid-throw the eccentric will thus require to be placed. By a moderate amount of attention to the action of the eccentrics, considered as cranks, and of the consequent movements of the valve, even a beginner in the study of the Locomotive may soon learn the principal details of valve setting—a matter often invested with much mystery in the mind of the tyro.

It is next necessary to describe the mode of connecting the valves with the eccentrics. Of the latter, there is one fixed to drive the valve of each cylinder, in forward gear, and another for backing. And it is necessary to be able to connect the valves either with the forward or the backward eccentrics in an instant. The respective eccentric rods, therefore, for working each valve (there being two eccentric rods to each valve, or four in all) are connected by turning joints, like those of a pair of nut-crackers, to the opposite ends of a piece of metal called the link, because it links the eccentric rods together. Thus the opposite ends of the link have the to-and-fro movements of the eccentrics with which they are respectively connected, so that the top of the link may be moving forward while the bottom is being drawn back, or *vice versâ*. It is usual to curve the links to the radius of the eccentric rods, as will be seen in most of the plates in the present work; but the engine shown in Plate III., as that also in Plates XVI. and XVII., has Allan's straight link. The link may be described as a thick iron plate, having either a curved or straight opening or slot (the latter only in Allan's link) cut completely through its thickness, this slot extending for nearly the whole length of the link. A rod connected to the valve spindle, and which, extending from the link towards the slide-valve chest, can easily be followed in Plate III., is forked at its end to grasp a block which fits closely, but not tightly, against the inner surfaces of the slot in the link. Now, if the link be raised or lowered upon this block, or if the link be supported at any fixed distance above the rails, and the block be raised or lowered in the slot of the link, or if, as in the case of Allan's straight link, the block is raised while the link is lowered, and *vice versâ*, the block, and consequently the valve rod and slide valve, will in each of these cases receive the movement of whichever eccentric rod it is thus brought in conjunction with. It will be understood that this raising and lowering of the link or the block, or both, as the case may be, is not a part of the motion of the engine, but is to be effected by the hand of the engineman whenever he desires to change the valves from forward to back gear, or the contrary, or whenever be wishes to vary the rate of expansive working, as will be explained further on. Beneath the link in the longitudinal section in Plate III. will be seen a shaft, in section (and marked as 3 feet 11 inches horizontally in front of the driving axle, and 1 foot 7½ inches below the centre line of the engine). Projecting from this shaft, in nearly opposite directions, are two short arms, the shorter of which is connected to the top of the link by a rod or bar which is jointed at each end in the same manner as the link is jointed to the eccentric rods. The link is supported by this jointed bar; and from the longer arm of the shaft, called the weigh shaft, another jointed bar supports the valve rod. The weigh shaft has an arm rising at one end of it, and which arm may be thrown forward or backward by a rod worked by a lever on the engineman's foot-plate. By this lever, then, the engineman can place the blocks on the ends of the valve rods in conjunction with either the forward or the backward eccentric rods; or he can place the blocks in any intermediate position between the ends of the respective eccentric rods. As the forward and backward eccentrics move in nearly opposite directions, the link has but little motion at the middle of its length; and when the block on the end of the valve rod is made to approach this part of the link, it receives a correspondingly smaller amount of motion than when it is opposite the end of either eccentric rod. Advantage is taken of this to work the steam expansively in the cylinders. For it has been seen that the valve is longer than the distance between the outer or extreme edges of the steam ports, and that, therefore, during a certain interval in each to-or-fro movement of the valve, steam is shut out from both ends of the cylinder, and the steam already in it is expanding as the piston moves forward, thereby giving out additional power beyond what would otherwise have been obtained from it. Now, if the distance by which the length of the valve exceeds that between the outer edges of the steam ports be 2 inches, it will be seen that in moving through this 2 inches, the valve must occupy a larger proportion of the time required to make its journey in either direction, when the whole movement is but 2½ inches, than it would do when the whole movement is 5 inches; and whatever may be the distance through which the valve moves, it will make this movement while the wheel is turning half round, or in the time occupied by the piston in making a single stroke. If, therefore, the whole movement of the valve be but 2¼ inches or 2½ inches, there will be a great proportion of the whole time, in which the piston is making its stroke, during which no steam will be admitted upon it, as the valve cannot admit steam while moving through its total overlap, which in this case is taken as 2 inches. But if the total movement of the valve, by being connected directly with the end of the link, and thus receiving the full motion of the eccentric, be say 5 inches, then the proportion of the whole time of a stroke of the piston during which the valve is moving through the total overlap of 2 inches will not be great, and hence the steam will be acting with its full pressure upon the piston during the greater proportion of its stroke. It is thus, by giving overlap or lap on the slide valve, as already explained, and by providing means for moving the valve with any required total amount of motion or "travel," that the steam may be cut off and worked expansively at different parts of the stroke of the piston. The student taking up the study of the Locomotive for the first time, may now refer to the general description of the action of the link motion on pages 60-65, and to the chapter which follows the present, for further and more precise information as to the valve gear.

We have now shown how the heat is applied to the water in a locomotive boiler to generate steam, and how the steam is applied to produce rotary motion in the engine. We have first seen how the fire (urged by an artificial draught of air, maintained by the escape of the steam after it has been worked in the engine itself) is surrounded everywhere by thin plates of metal separating it from the water, and how the heat is strained, so to speak, through small tubes, until nearly all of it that is available is taken up in the generation of steam. We have seen how the steam is accumulated in the upper part of the boiler, and the means employed to prevent its pressure from becoming too great, and the means for ascertaining that the water is maintained at its proper level. We have followed the steam through the starting valve, called the regulator (although the engine is now regulated almost entirely by the link motion), and down to the slide-valve box. Thence we have traced it past one or the other edge of the slide valve into the corresponding end of the cylinder, there to act upon the piston. We have followed the to-and-fro motion of the piston, and its conversion into the rotary motion of the crank; and we have traced the action of the eccentrics, turning with the cranks, in working the slide valve to each cylinder, so as to admit and release the steam at the right moment. The description has been lengthy, no doubt; but the Author is aware, from his own experience, that even a redundancy of words in mechanical descriptions is better than too few. In the foregoing description it has been the object, also, to give reasons for what is described. As a rule, students cannot, when new to the subject, perceive these reasons for themselves; and it is not uncommon to hear those, whose wish it is to master the Locomotive fully, complain of the scantiness and poverty of the descriptive matter usually found in the books. The Author has himself read nearly every work ever published upon the Locomotive Engine; and yet he is compelled to say that he had to discover for himself the *reasons why* tube surface, beyond a certain limit, adds nothing to the efficiency of the fuel; *why* the hot air is carried up the chimney by the steam blast; *why* the rate of expansion is varied by varying the travel of a lap valve; *why* and how the steam acts directly upon the cylinder covers as well as upon the pistons in moving the engine along; *why* the wear on the front and back sides of the axle boxes is unequal; *why* the pressure is almost constantly upward at the cross-heads, &c. Not, by any means, that he was the first to discover any one of these things in themselves, for they must all have been known to others, and the reasons are indeed obvious enough upon a little reflection. But the books to which the Author resorted did not, as a rule, contain any of these reasons. They described, rather, how the Locomotive Engine was constructed, but did not explain clearly why it was thus constructed, nor, in all cases, the real action of the parts.

Having the boiler and working machinery, its support upon a framing with wheels and springs is easily understood. The wheels are fixed securely upon the axles, and the latter revolve in "boxes," upon which the weight of the boiler and machinery is carried through stout springs. In Plate III. the framing will be clearly seen in the "plan." It consists of four bars, extending the whole length of the engine, and connected together at the ends, and at the middle, beneath the boiler. These bars, 12 inches deep and 1 inch thick, have great stiffness in a vertical direction, and the inner pair (marked 4 feet 11 inches apart in the plan) carry the whole weight of the boiler, which rests upon them at the fire box and smoke box. The outer frame bars (marked 7 feet 2 inches apart) are strongly secured to the inner bars by iron plates and angle iron. The inner or main frame bars receive the whole of the to-and-fro thrust and pull of the steam in the cylinders, which are strongly bolted between them, the cylinders being firmly fastened also to each other by bolts, as shown in the "section through cylinders," Plate IV. It will be understood that the slide-valve chest, instead of being a separate part, is in this case formed one-half with each cylinder.

The frame bars are so formed (and the form can be better seen in the engravings than described in words) that the axle boxes can rise and fall freely so far as the springs, which rest upon them, and which springs are secured at their ends to the frame, will permit. In the engine which has been described the axle boxes of the driving axle are guided vertically by suitable guides, or axle guards, in the inner pair of frame bars. The axles of the leading and trailing wheels are extended, however, beyond their wheels, and revolve within boxes guided by the outer pair of frame bars. That part of the axle which revolves in contact with the axle box is called the journal, and we here speak of the inside journals of the driving axle, and of the outside journals of the leading and trailing axles, because the journals are respectively inside and outside of the wheels. The springs, which are seen in dotted lines and in outline in the longitudinal section, Plate III., are formed of steel plates, from ⅜ inch to ½ inch in thickness, and of a number proportionate to the load to be carried, each spring of the driving axle having often to carry from 4 to 6 tons.

The wheels, almost always of wrought iron, except in American engines, are encircled by tyres of either the best wrought iron or steel. They are kept upon the rails by a flange formed upon them, as shown in the various sections of the tyres in Plate IV. As long as the wheels are fast upon the axle, and the rails upon which they run are maintained at the right distance apart, neither wheel (except there be some obstruction) can leave the rails, for the flange of either wheel prevents its going outward over and beyond the rail, and the flange of its fellow wheel, on the other end of the axle, prevents it from dropping within the rail.

The tender hardly requires a detailed description. It is a carriage on four or more wheels, and having a plate-iron tank for conveying water, and space, also, for coal, coke, or whatever may be the fuel to be burnt. The floor of the tender is generally at the same level as the foot-plate upon which the engineman stands, close to the fire box, and where he commands a view of whatever may be in front. By means of levers, variously arranged in different tenders, thick blocks of wood, or blocks faced with iron or leather, may be forced against the peripheries of the wheels of the tender, so as, when it is desired to stop, to check their revolution, in the same manner as would the brake of an ordinary carriage in descending a steep hill.

In looking over the engravings of different engines illustrated in the present work, it will be seen that very considerable differences of arrangement are to be met with in different examples. Plate VI. is what may be called a skeleton drawing, as compared with the clear and elaborate detail of Plates III. and IV. A beginner would not obtain a clear idea of the construction of a locomotive engine from Plate VI., yet it represents the most useful class of drawings for the purposes of the working engineer. The student, too, can by this time understand enough of it to comprehend the many differences between it and the express engine already described. In the first place, the engine shown in Plate VI. is intended for a very different kind of work—that of drawing goods trains, at a slow speed, up a long incline, in some places rising as steeply as 1 in 37. Consequently the cylinders are much larger, the stroke of the piston is greater, and the driving wheels are much smaller than in the engine shown in Plate III. If we suppose the same pressure of steam per square inch in each case, say 100 lbs., then the 20-inch pistons of the goods engine will sustain more than one-half as much more total pressure than the 16-inch pistons of the express engine; for the area of a 16-inch piston is 201 square inches, while that of a 20-inch piston is 314 square inches. And as the longer the crank and the shorter the radius of the driving wheel, the greater the proportion of the pressure on the pistons which will be exerted as tractive force at the rail, we have a 12-inch crank, and a 2-feet 2-inch radius of driving wheel in the goods engine, as compared with an 11-inch crank and a 3-feet 6-inch radius in the express engine. With a pressure of steam of 100 lbs. per square inch upon the pistons in each case, the mean tractive force of the express engine would be 6,705 lbs., while that of the goods engine would be 18,461 lbs. These numbers may be taken, therefore, as representing the relative weights of the trains which the two engines could respectively draw upon the same line of railway; and it happens, in this case, that both engines are made for the 5-feet 6-inch gauge of rails, the "gauge" being the distance between the inner edges of the rails of a line of way. But if the supply of steam, of the given pressure, was in each case sufficient to keep the pistons in motion at the same speed, say 500 feet per minute, the express engine would move the faster with its load; for not only would it advance 22 feet in making a single revolution of its 7-feet wheels, where the goods engine would advance but 13 feet 7 inches at each turn of its 4-feet 4-inch wheels, but the pistons of the express engine would each move but 3 feet 8 inches in making a double stroke, or a single revolution of the driving wheels, while the pistons of the goods engine would each move 4 feet in turning the smaller wheels once round. In practice, the difference in the loads drawn and in the speed maintained would be greater than this comparison would show: for the express engine never maintains an average pressure of 100 lbs. per square inch upon the pistons, although the goods engine often does, and the pistons of the express engine generally move at the mean rate of from 800 to 100 feet per minute, while those of the goods engine move at from 300 to 500 feet. Hence it is that the express engine shown in Plate III. would probably take a train of 80 tons at 60 miles an hour on a level, while the goods engine would draw a train of 800 tons at say 15 miles an hour upon the same line.

But in order to exert great tractive force, the driving wheels of the engine must, by their friction upon the rails, have an "adhesion," as it is called, equal to this tractive force. Thus, to exert, as already calculated for the goods engine, 18,461 lbs. of tractive force, the driving wheels of the engine must not slip upon the rails, without advancing, until a resistance of at least 18,461 lbs. is opposed to them. This friction or "adhesion" is secured by providing sufficient weight upon the driving wheels. On a clean, dry rail as much as one-fourth, and even more, of the whole weight upon the driving wheels is available for adhesion. One-sixth of the weight is, however, a proportion more within the ordinary contingencies of weather, and of the general condition of the line. Taking one-fourth as nearly the utmost effective adhesion under any circumstances, the driving wheels of the goods engine should have 18,461 × 4 = 73,844 lbs., or nearly 33 tons upon them. This weight could not, however, be supported by the rails without injury, if applied through a single pair of wheels. Hence two or three pairs of wheels (in the goods engine shown in Plate VI., three pairs) are coupled together by coupling rods, so as to make them all driving wheels. It is customary to apply the term "driving wheels" to the single pair only which directly receive motion from the pistons; but although these wheels may be said to drive those coupled to them, all the coupled wheels act together in driving the engine. In Plate VI. the three pairs of wheels each 4 feet 4 inches in diameter, and coupled by cranks and coupling rods, as shown clearly at the bottom of the plate, press with a weight of from 33 to 35 tons upon the rails, each single wheel bearing with a weight of nearly 6 tons. The united adhesion of the six wheels—and neither can turn unless all turn together—enables an extreme tractive force of even 18,461 lbs., as already calculated, to be exerted. The whole engine is estimated to weigh about 48 tons in working trim, a portion being supported upon a swivelling bogie of four wheels, each 2 feet 9 inches in diameter in front. It will be understood that the goods engine shown has no separate tender, but that 1,050 gallons, or 4 tons 13 cwt. 3 qrs. of water are carried in a saddle tank over the boiler, besides perhaps a ton or so of coal on the foot-plate.

The object in employing coupled driving wheels is simply to distribute the great weight necessary for adhesion, where great tractive force is to be exerted at moderate speeds. In the engine shown in Plate V. the whole weight is supported upon three pairs of wheels, all of the same diameter, and coupled together. This engine is of the usual type of English goods engines for the older lines.

Referring again to Plate VI., it will be seen that the fire grate, instead of being horizontal, is considerably inclined, as is that also of the engine in Plate V., and the fire box is furthermore divided, for the greater portion of its length, by a vertical water space, or "mid-feather," so as to form two compartments of the fire box, communicating with each other by an opening in the mid-feather near the tube plate. Thus there are two separate grates, each fed through its own fire door. The double inclined grate, each half being fired alternately, was first adopted in general use by Mr. Cudworth, of the South-Eastern Railway, for his coal-burning engines, and it is now in extended use. In other respects the boiler is much the same as, although larger than, that of the engine already described in connection with Plates III. and IV. In order that the connecting rods, cross-heads, &c., may work clear over the axle of the front pair of driving wheels, the cylinders, instead of being placed horizontally, are inclined, in the case of Plate VI., at an angle of 1 in 8½. The framing is double, or in four parallel longitudinal members, as in Plate III., and the outer framing is made of double plates of iron, with hard wood rivetted between them. This is an old construction of framing now generally abandoned. The principal pair of driving wheels have no flanges, the engine being kept upon the rails by flanges upon eight of the whole number (ten) of its wheels. The axle of the principal pair of driving wheels revolves within four axle boxes, one on each side of each driving wheel; these axle boxes being guided respectively in axle guards in the outside and inside frames. The axle boxes of the front and back driving wheels are formed only in the outer framing. The same general arrangement of framing and axle boxes is shown still more plainly in Plate V. The engine shown in Plate VI. possesses other features differing from those of the engine shown in Plates III. and IV., but these may be better seen in some of the other plates.

Plates I. and II. illustrate an engine, the favourite pattern on the London and North-Western Railway, in which the cylinders are placed differently from those of either of the engines already described. Instead of being placed within or directly beneath the smoke box, they are placed outside of it, and at such a distance apart that their connecting rods, instead of working upon cranks formed in the driving axle, work upon crank pins fixed in the wheel itself, as clearly shown in the "section at crank," in the lower right-hand corner of Plate II., where one of the crank pins, 4 inches in diameter, is fixed in the wheel at a distance of 12 inches above the centre of the driving axle, thus corresponding to a 24-inch stroke of the piston. The cylinder is not quite horizontal, but is slightly inclined, the object being to thus place the leading wheels a little further forward, so as to prevent so much weight from overhanging in front as would be the case with horizontal cylinders. The arrangement of cylinders outside of the smoke box possesses several advantages over that of cylinders within the smoke box, as in Plate III., and known as "inside cylinders." In the first place, the costly and heavy crank axle of the inside-cylinder engine is dispensed with. The crank axle, besides being heavy and costly, is inherently weak, and is always liable to break—so much so that the London and North-Western Company could only secure themselves against the risk of frequent breakage by resorting to cast-steel crank axles, at a cost of £250 each. For the same reason, among others, only outside-cylinder engines have been made for some years by most of the continental makers, and no inside-cylinder engines are made in America. Again, with inside cylinders considerable room is required beneath the boiler, in order that the cranks may turn without striking. This room is not required, and is therefore saved in the case of outside-cylinder engines. A good illustration of the extent of this saving is afforded in a comparison of two of the London and North-Western Railway engines in the International Exhibition of 1862. One was an inside-cylinder engine, with 7-feet 6-inch driving wheels; the other, the outside-cylinder engine shown in Plates I. and II., with 7-feet 7½-inch driving wheels. The centre of the cylindrical barrel of the boiler of the inside-cylinder engine was at a distance of 7 feet 5½ inches above the rails of the line; while the centre of the boiler of the outside-cylinder engine, although the latter had rather larger wheels, was but 6 feet 6½ inches above the rails. This comparison applies more especially to the narrow, or 4-feet 8½-inch gauge. On a wider gauge, inside cylinders would be placed so far apart that the cranks might clear the boiler, even when the lower part of its cylindrical barrel was near the driving axle. There is the advantage, too, with outside cylinders, that all, or nearly all, the machinery is exposed to the eye of the engineman, even when the engine is working; and it is all easily accessible for repairs. On the other hand, the centres of the cylinders are 6 feet apart in the engine shown in Plates I. and II.; and if there be the least "slack" in the driving axle boxes, the engine is more readily swerved, at each stroke of the pistons, to the right or left, than if the cylinders are placed nearer to each other. Especially if the weights of the pistons, piston rods, cross-heads, and connecting rods, are not counterweighted in the driving wheels, the to-and-fro thrust, due to the momentum of these weights, tells much more, in the way of alternate right and left motion of the engine across the line, in the case of outside than of inside cylinders. It has already been explained, too, that in running forward, the cross-heads press constantly upwards upon the guide bars, except only when the pistons are at the ends of their strokes. If the pistons are 6 feet apart, this alternate upward pressure upon opposite sides of the engine tends to cause a rolling motion from side to side, and to a greater extent than where the pistons are, say but 2 feet apart, as measured across the width of the engine. Outside-cylinder engines, therefore, require to have their axle boxes kept in good fitting to the axle guards: they require to have all the reciprocating parts connected with the crank pins properly counterweighted in the driving wheels; and they also require to have stiff springs to the leading wheels, to prevent the rocking from side to side which would otherwise result from the alternate upward pressure upon the guides of the cross-heads. These requisites for outside-cylinder engines are now generally fulfilled, and they are, in consequence, nearly as steady, when running at great speed, as inside-cylinder engines. It has been argued that outside-cylinder engines, from the exposed position of their cylinders to the air, waste more of their steam by condensation than do inside cylinders, which are partly immersed in the hot gases of the smoke box. It is necessary only, however, to cover outside cylinders thoroughly from the cooling action of the air; and in American engines the outside cylinders everywhere employed there are encased with brightly polished brass coverings, so as to radiate as little heat as may be. Thus protected, no considerable loss ensues from the condensation of the steam within the cylinder. Altogether, or after all points in favour of and against them are considered, outside-cylinder engines possess a decided advantage over those with inside cylinders; and it is quite likely that the former will before many years entirely supersede the latter in this country, as they have already nearly done on the Continent and in America.

Plates VII. and VIII., together with Plate XIV., illustrate an outside-cylinder express engine of great size, as made for the Caledonian Railway, and exhibited in the International Exhibition of 1862. Here the cylinders, 17¼ inches in diameter, and for a 24-inch stroke of piston, are laid horizontally, and although the driving wheels are of the great diameter of 8 feet 2 inches, the centre of the boiler is but 6 feet 6¾ inches above the rails. On referring to Plate XIV., however, it will be seen that the boiler is indented on its lower side, to permit the eccentrics to clear it, and the link is of the "stationary" kind, the valve rod being made to rise and lower as required. Were the rising and falling link employed, it would, when in back gear, strike the bottom of the boiler. Plates IX. and X. represent the large outside-cylinder express engines of the Great Eastern Railway. Here the cylinders are inclined from the horizontal, but only enough to gain a little room in order to place the leading wheels further forward. The fire grate is inclined, and the boiler differs in points of minor detail, but not in principle, from those previously shown. Plates XVI. and XVII. show the still larger express engines of the London and South-Western Railway. The chief peculiarity of these engines is in their boilers for burning coal. The construction is best shown in the longitudinal section, Plate XVI. The fire box is divided into a front and a back compartment by a sloping water-space, or mid-feather, extending from about the middle of the length of the fire grate upwards and backwards towards the doors, of which there is one to each compartment, one door being placed over the other. The fire is maintained chiefly upon the back division of the fire grate, which has an ash pan and damper separate from that pertaining to the front division. The damper of the ash pan to the front grate is kept closed, or nearly so, and only a little coal is burnt upon the front grate. The flame and gases from the back grate rise through tubes in the inclined water-space, and between a fire-brick grating at the top of this space, into the front and larger compartment of the fire box. Fresh air is admitted to the gases through a number of hollow stay bolts in the sides of the fire box. The inflaming gaseous volume is then deflected downwards by a hanging water-space depending from the fire-box crown at the middle of its length. Thence, mixed with more fresh air, it passes through a series of tubes, only a few inches long, into a combustion chamber, 4 feet 6 inches long, within the barrel of the boiler, and thence the hot air and gases of combustion escape to the smoke box through a series of more than 375 small tubes, about 5 feet 4 inches long, these tubes being of about double the number and of half the length of those employed in other engines of the ordinary construction. The object of the whole arrangement is to bring the gas and air into thorough mixture, so as to insure the thorough combustion of the former. Boilers of this construction, although costly and heavy, have been in use, with good results, for about eight years on the London and South-Western and other lines. In the London and South-Western and the South-Eastern express engines there are two pairs of coupled driving wheels to obtain increased adhesion without excessive weight upon a single pair of wheels. In the engine shown in Plates VII. and VIII. the weight upon the single pair of driving wheels is 14 tons 11 cwt., which is greater than ordinary iron rails can well bear.

The present chapter will be the most suitable for giving a few general dimensions of engines of average size and power. It must be kept in mind that, despite all the formulæ occasionally to be met with in certain treatises, the proportions of locomotive engines are seldom, if ever, determined by exact rules; and the student will, sooner or later, have to observe and compare the dimensions and proportions fixed by experience, which is the foundation of all sound knowledge in locomotive engineering. The dimensions of an engine for the common or 4-feet 8½-inch gauge, will be the most useful, and we will give, therefore, those of the express engine shown in Plates I. and II. This engine has 16-inch cylinders, 24-inch stroke, and a single pair of 7-feet 7½-inch driving wheels. The leading and trailing wheels are each 3 feet 7½ inches, the horizontal distance from the centre of the driving to the centre of the leading wheel being 7 feet 7 inches, and from the driving to the centre of the trailing wheel 7 feet 10 inches, making a total wheel-base, or length covered by the wheels upon the rails, of 15 feet 5 inches. The inside fire box has a mean length of 4 feet 1½ inches, being a little more at the bottom and a little less at the top. Its width is 3 feet 6 inches, and the height of the crown-plate above the fire grate is 5 feet 4¼ inches. The area of the fire grate is 14·9 square feet. The barrel of the boiler is formed of three rings of plates, arranged telescopically, the mean internal diameter being 3 feet 10¾ inches. The centre of the cylindrical barrel is 6 feet 6¾ inches above the rails. It contains 192 tubes of brass, 1⅝ inches in external diameter, 2⅛ inches apart from centre to centre, and of a length of 10 feet 9 inches between the tube plates. The internal heating surface of the fire box is 85 square feet, and the external surface of all the tubes is 1,013 square feet. The iron plates of the boiler are ⁷⁄₁₆ inch thick, the copper plates of the fire box ½ inch, except at the tube plate, which is thickened to ¹³⁄₁₆ inch. The copper stay bolts connecting the inner and outer fire boxes are ⅞ inch in diameter, and 4 inches from centre to centre, both vertically and lengthwise of the fire box. The chimney is 15½ inches in internal diameter, and the height of the top of the chimney from the rail 13 feet 1 inch. The pair of safety valves are each 3 inches in diameter.

The steam pipe is 5½ inches in diameter internally, and the copper branch steam pipe to each cylinder 3¾ inches. The steam and exhaust ports are 13½ inches long, the steam ports being 1½ inches wide, the exhaust port 3¼ inches, and the bridges between the ports 1⅜ inches. The slide valves, when placed in the middle of their throw, so as to cover both steam ports equally, have ¾ inch lap at each end, and say ¹⁄₁₆ inch inside lap. When in full gear the valve has ¼ inch "lead;" or, in other words, the steam port is already ¼ inch open when the piston is at the end of its stroke. The greatest movement of the valve to and fro in full gear is 3⅝ inches. The valve face is 1 foot 3½ inches from the centre line of the cylinder; and when viewed from one side, the centre of the valve spindle is 2 inches below the inclined axis of the cylinder. The orifice of the blast pipe is 4¼ inches in diameter, and 9 inches above the centre of the barrel of the boiler; and it is accurately adjusted so as to discharge the waste steam exactly up the centre of the chimney.

The cylinders are 6 feet apart at centres, as measured across the engine. They are ¾ inch thick, and are fastened by a broad flange and by bolts to the frame. The piston is packed, so as to be steam-tight, by three rings of steel wire, each ¼ inch square, according to Mr. Ramsbottom's patent. The piston rod, of steel, is 2½ inches in diameter. The piston rod has a bearing of 3½ inches in the boss of the piston, and of 4½ inches in the boss of the cross-head. The connecting rods are 6 feet 2 inches long from centre to centre of brasses; they are 2¾ inches deep and 1⅝ inches thick next to the cross-head, and 3¾ inches deep and 2 inches thick next to the larger end. The pair of cross-head pins in each cross-head are 2¼ inches in diameter and 2 inches long; and the crank pins in the wheels are each 4 inches in diameter and 4 inches long. The crank pins are 4½ inches in diameter where they are fixed in the wheels, and their bearing in each driving wheel is 5½ inches long. The driving wheels, 7 feet 3 inches inside the tyre, or 7 feet 7½ inches over all, are keyed each by a single key only to the axle. The axle is 6¼ inches in diameter in the body; the journals are 7 inches in diameter and 7 inches long; and those parts of the axle passing through the wheels are each 6½ inches long and 8½ inches in diameter. The leading and trailing axles are each 5¼ inches in diameter in the body, the journals 6 inches in diameter and 10 inches long, and those parts of the axle to which the wheels are fixed are 7¼ inches in diameter and 6½ inches long. It should be said here that these proportions of the various parts of engine axles differ from the usual practice, and that they are those adopted on the North-Western Railway. It is argued, and with reason, by Mr. Ramsbottom, that if the weakest part of the axle be between the journals, it will, in case it does break, first break there, and that the wheel will be still kept upon the rails by the journal, grasped by its axle box, still held securely in the framing.

The eccentrics, of 5 inches total motion, or 2½ inches throw, are 13 inches in diameter, and 2⅝ inches thick, as measured along the axle. They are connected, by rods 3 feet 10 inches long from the centre of the eccentric to the centre of the link, with a slotted link of 3 feet 10 inches radius, the rods connecting with the ends of the link at a distance there of 16 inches apart. The links, one for each pair of eccentrics, work the valves of the respective cylinders. The dimensions of the valves and steam ports have been already given. The exhaust steam pipes from the two cylinders are united in a single pipe, which is directed upwards directly in the centre line of the chimney, the open orifice of this single pipe being 4½ inches in internal diameter, and placed at an elevation of 9 inches above the centre line of the boiler.

The parallel bars of the main or inside framing are 1 inch thick, and of variable depth, nowhere less than 11 inches, and they are 4 feet 2 inches apart in the clear. The shape of the framing is such as to permit the longitudinal strains caused by the pressure of the steam in the cylinders to be transmitted nearly in a line corresponding to the axes of the main bars of the framing, these bars being bent down at each side of the axle guard of the driving axle, as shown. The axle guards are forged or cast separately, and bolted to the framing bars, and the latter are strongly connected by cross-braces, as well as by the buffer beam, and a stout bar at the hind end. The shape and dimensions of the driving and leading springs are shown, the trailing axle being made to carry its load through six volute springs. The volute form is employed in this case merely because it occupies but little room in the direction of the length of the framing. Were the ordinary form of springs employed, they would extend some distance along the sides of the fire box, and as the clear distance, transversely, between the tyres of each pair of wheels is fixed, say 4 feet 5½ inches for the narrow gauge, room for the springs could only be had by making the fire box of less than its present width.

All these, as well as other dimensions of the engine, are governed by reasons founded in experience, and in but few cases can these dimensions be traced to any general law. Were the engine intended for a higher average speed than that at which it is commonly worked, it would require a larger boiler, or, otherwise, its load would need to be diminished: for a slower rate of speed, on the contrary, a smaller boiler would suffice. Nor have the dimensions of the boiler any exact or even approximate relation to the quantity of water evaporated, except where the speed of the engine is considered. Thus, in the case of the engine shown in Plates I. and II., its boiler will, in express-train service, evaporate 150 cubic feet of water per hour. But in the case of a goods engine having a boiler of the same dimensions, and worked at say 15 miles an hour, it would not, however heavily worked, evaporate more than about 100 cubic feet of water per hour, with the same description of fuel and the same diameter of blast pipe. Again, the fire grate is of a good size where coke or free-burning coal is burnt, and it would indeed answer very well with wood. But were it attempted to burn "slack" coal, or screenings, as is the case on some of the French railways, a much larger grate would be found necessary, in order that sufficient air for combustion might find its way through the closely lying mass of burning fuel. Were a smaller driving wheel to be employed for the same service, either a smaller diameter of cylinder or a shorter stroke of piston would suffice, although, did the stroke remain the same as before, the pistons would move more rapidly, and possibly the working parts might show an increased tendency to heating. At the same time, the frequency with which the steam was exhausted from the cylinders would be increased; and it has been found that a given boiler will evaporate more water where the blast is comparatively frequent, although light, than where the puffs of steam in the chimney, although greater in volume, are separated by longer intervals of time.

The dimensions of the working parts are related to the quality of the materials employed. Where steel is employed in the place of wrought iron, a given strength is had, with less size and weight. The steel driving axle of the 30-ton engine shown in Plates IX. and X. is but 5½ inches in diameter in the journals, whereas an iron axle for a like engine should be 7 inches in diameter, having, therefore, upwards of half as much more metal in its cross section. The steel axles shown have broken, however, and are being replaced by larger axles of the same material. In other cases wrought-iron axles are preferred, as in the engine shown in Plates I. and II., in order that the journals may be case-hardened, whereby they are harder, and by so much more durable, than even steel itself of the kind employed for axles; for a steel axle must be of a quality known as "mild" and untempered, else it would be brittle, and break more readily than even inferior iron. In the class of engines shown in Plates I. and II., Bessemer steel is now employed for piston and connecting rods, guide bars (or "motion" bars, as they are often vaguely termed), and for the parts of the valve gear, excepting the links, which, when of case-hardened wrought iron, have all the toughness of the latter material, with the hardness of wearing surface of the hardest steel.

It has been a favourite practice with writers upon the Locomotive Engine to deal with it geometrically and with reference to its dimensions. For the useful purposes of railways, however, even more depends upon the excellence of its structure; and it may be said that the staunchness of the boiler, the hardness of the cylinders and valve faces, the strength of the cylinder fastenings, and the general soundness of the engine as a whole, have really more to do with its successful working than any abstract consideration of "heating surfaces," and many of the quantitative refinements so often insisted upon. Even the geometrical accuracy desirable in the valve gear is important rather from what experience has shown than from any precise demonstrable relation to the motion of steam. To this day successful locomotive engineering owes far more to practice than to theory; and it should never be forgotten that the best proportions of locomotive engines, so far as engineers are agreed upon which are best, have been discovered by practical men, with little or no aid from theory, except in so far as theory may in itself be considered as the generalization of successful practice.

The general description given for one kind of engine will serve for all, however different may be their visible forms. Locomotive engines have been made with one, two, three, four, and even five steam cylinders, yet the general construction of the boiler, and the action of the steam in the cylinders is the same in all. There is always the reciprocating piston, giving rotary motion to one or more pairs of wheels, and the valve gear must always give, practically, the distribution and discharge of the steam already described. Those who have experienced a love for the study of mechanical engineering may derive almost any amount of gratification from the study of the Locomotive Engine, by learning the *reasons* which govern any specific mode of construction, and not less the reasons representing the defects of a different mode, than those which may directly support any form or arrangement actually employed.

CHAPTER VI.

VALVE GEAR CONSIDERED GEOMETRICALLY;[*] THE SLIDE VALVE AND ITS MOTION WHEN DRIVEN BY A SINGLE ECCENTRIC.

The manner in which the steam is admitted to, or released from, the cylinder of a steam engine, and the points of the stroke of the piston at which these events commence and terminate are subject to variation in three distinct ways, —*Firstly*, By an alteration in the form or proportions of the valve or valves by which the distribution of the steam is effected; *Secondly*, By a variation in the valve gear driving such valve or valves; and, *Thirdly*, By an alteration in the relative proportions of the connecting rod and crank. The action of each of these varying causes may be shown either by a model of the particular arrangement under consideration, or by the method of geometrical construction, the latter being generally preferable for the purpose of investigation.

In this and the following chapters it is intended to explain the manner in which such an investigation can be carried out; and in order to do this, it will be convenient that the subject should be divided into three parts, as follow:—

1*st. The action of an ordinary slide valve driven by a single eccentric.*

2*d. "Link motions," commonly so called.*

3*d. Expansion gears not comprised under the preceding head.*

Each of these divisions will be considered separately in the order above named.

The action of the ordinary slide valve and single eccentric. Definitions.—The form of valve generally employed is that shown by Fig. 122, which will serve to explain the designations applied to the various parts of the valve and of the face upon which it works. In this latter are formed openings, or *ports*, of which there are generally three, the two outer ones (A A) being the *steam ports*, or entrances to the passages by which the ends of the cylinder are connected with the steam chest, and the centre one (B), the *exhaust port*, affording communication with the exhaust pipe. Those portions of the face (C C) between the ports are termed *bars* or *bridges*. In the figure the valve is represented in the position it would occupy when at half-stroke; and the amount (D) by which, when so situated, the face of the valve extends at each end beyond the steam ports is called *outside lap* or *outside cover*, or more frequently, simply *lap*. On the other hand, the distance (E) to which, while the valve is still at half-stroke, its face extends within the inner edge of the steam ports, is termed *inside lap* or *inside cover*. Sometimes the valve is formed as shown by the dotted lines, so that when in this central position its face does not quite cover the steam ports on the inside, but leaves them both partially open to the exhaust. The amount (F) thus left uncovered is denominated *inside clearance*. The distance to which, at the commencement of the stroke of the piston, one of the steam ports is open externally for the admission of steam is termed the *lead on the steam side*, or *outside lead;* while the extent to which the other steam port is at the same time open to the exhaust is called *inside lead*, or *exhaust lead*. In practice, *outside lead* is most generally denominated simply *lead*.

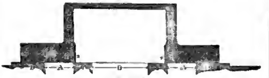

Fig. 122.

In order to express the proportions and position of the eccentric the following terms are very generally employed:—

The *throw* (A) (see Fig. 123) of an eccentric is the amount of its eccentricity, or the radius of the circle described by its centre during a revolution of the crank shaft.

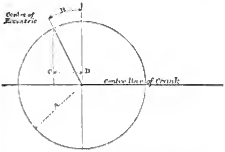

Fig. 123.

The *angular advance* (B) is the angular measurement of the arc described by the centre of the eccentric while passing from the position which it occupies, when the valve is at half-stroke, to that which it assumes at the commencement of the stroke of the piston.

Linear advance (C D) is the distance which the valve, if coupled direct to the eccentric, will travel while the eccentric is moving through the arc forming its angular advance. This linear advance is, as will be presently seen, equal to the sum of the outside lap and lead, to the sum of the inside lap and lead, or to the difference of the inside lead and clearance.

When a slide valve is driven by a single eccentric its motion is derived from two sources:—*First*, There is that given by the rotation of the centre of the eccentric round that of the crank shaft; and, *Secondly*, There is an acceleration and retardation of the motion so given, caused by the varying obliquity of the eccentric rod. As the motion derived from an eccentric is precisely similar to that obtained from a crank of the same throw (the former being in fact merely a crank with the diameter of the crank pin excessively increased), it follows that the position of the piston corresponding to any point in the revolution of the crank shaft is also determined by the action of the crank and connecting rod conjointly.

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[*] This chapter has been prepared, at the Author's request, by his friend, Mr. W. H. Maw, of the Great Eastern Railway.

The manner in which the position of the piston is influenced by the action of the connecting rod is shown by Fig. 124, in which are represented the relative positions occupied by the cross-head and crank, at nine points in a half revolution of the crank shaft. In this diagram the length of the crank is taken at 12 inches, and that of the connecting rod as 6 feet. The short lines, numbered 1 to 9 on the upper side of the centre line, represent the positions of the cross-head corresponding to the similarly numbered positions of the crank. The other lines below the centre line, lettered *a* to *i*, show the places the cross-head would occupy if the connecting rod was of infinite length; and the spaces into which these lines divide the stroke of course agree with the spaces into which the diameter, *k l*, is divided by the ordinates drawn to it from the points denoting the positions of the crank pin. This diagram at once shows that if the crank shaft rotates in the direction of the arrow, from *k to l*, the motion of the piston will, during the first half of the stroke, be retarded by the action of the connecting rod, while during the latter half it will be correspondingly accelerated. The effect of this is, that during the whole of the stroke *from* the crank shaft the piston is in arrear of the position which it would occupy if the connecting rod was of infinite length, whilst during the stroke *towards* the crank shaft it is correspondingly in advance of such position. In the example before us the piston travels 10·99 inches, while the crank moves from position 1 to that marked 5, and 13·01 inches as it rotates from 5 to 9.

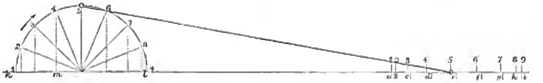

Fig. 124.—Diagram showing the Influence of the Connecting Rod on the Position of the Piston.

It is evident that the motion of the piston during the two halves of the stroke becomes more and more symmetrical as the length of the connecting rod, considered relatively to the throw of the crank, is increased, and, as the length of the eccentric rod always bears a much greater proportion to the throw of the eccentric than the length of the connecting rod does to that of the crank, it follows that the motion of the slide valve is much less irregular than that of the piston. With the proportions of eccentric and rod usually employed the disturbance arising from the cause above discussed is, when the valve is coupled direct to the eccentric, practically inappreciable; and in most of the following elementary investigations of the action of the slide valve it will therefore be neglected, and we shall thus be able to determine the motion of the valve by the method of ordinates, shown on the diagram explaining the action of the connecting rod and crank.

The amount of disturbance produced by the action of a connecting rod bearing any given proportion to the throw of the crank may be easily ascertained by a simple trigonometrical formula,* and a table is subjoined containing the alterations in the positions of the piston, produced by connecting rods five, six, and seven times the length of the crank, which have been calculated in this manner.

TABLE OF VARIATIONS IN THE POSITION OF THE PISTON CAUSED BY THE ANGULARITY OF THE CONNECTING ROD.

Distance of Piston from the Commencement of the Stroke, due to the Linear Progress of the Crank, and Measured in per Centages of the whole Stroke.		Corrections, in per Centages of the Stroke, due to the Angularity of the Connecting Rod; the Length of the Rod being		
		Five times the Length of Crank.	Six times the Length of Crank.	Seven times the Length of Crank.
0	100	0·	0·	0·
2	98	0·39	0·33	0·27
4	96	0·77	0·65	0·55
6	94	1·14	0·95	0·81
8	92	1·48	1·23	1·05
10	90	1·81	1·5	1·29
12	88	2·13	1·77	1·52
14	86	2·43	2·02	1·73
16	84	2·7	2·25	1·93
18	82	2·97	2·47	2·12
20	80	3·23	2·68	2·3
22	78	3·47	2·88	2·46
24	76	3·68	3·06	2·62
26	74	3·88	3·23	2·76
28	72	4·07	3·38	2·89
30	70	4·24	3·52	3·02
32	68	4·4	3·65	3·13
34	66	4·53	3·77	3·23
36	64	4·66	3·87	3·31
38	62	4·76	3·96	3·39
40	60	4·85	4·03	3·45
42	58	4·92	4·09	3·5
44	56	4·98	4·14	3·54
46	54	5·02	4·18	3·57
48	52	5·04	4·19	3·58
50	50	5·05	4·20	3·59

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* The formula by which the accompanying table was constructed is the following:

$$c = 50\,n - \sqrt{2{,}500\,(n^2 - 1) + d^2}$$

in which c = the amount of the correction in *per centages* of the stroke; d = the uncorrected distance of the piston from the *middle* of the stroke, also expressed in *per centages* of the stroke; and n = the number of times that the length of the connecting rod contains that of the crank. An approximate correction, sufficiently accurate for most practical purposes, may also be found by the following more simple rule:—Multiply together the two distances into which the uncorrected position of the piston divides the stroke, and divide the product by twice the length of the connecting rod—the quotient will be the correction required. In using this rule the lengths of the connecting rod and of the two portions of the stroke must be expressed in terms of the same unit of measurement.

The corrections in the above table are *additive* for strokes *towards*, and *subtractive* for strokes *from* the crank shaft: the manner in which they are to be used will be illustrated by the following example:—Let the crank in Fig. 124, revolving in the direction of the arrow, have attained a position (4) such that an ordinate drawn from the centre of the crank pin to the centre line of the engine cuts the latter at a point, *m*, and let the distance, *k m*, equal 30 per cent. of the whole diameter. Then, if the piston is connected by a rod, the length of which equals five times the throw of the crank, it will have travelled 30−4·24=25·76 per cent. of its stroke when the crank has arrived at the position indicated; 4·24 being the correction found in the table, in the column headed "Five times the Length of Crank," and opposite the number 30 in the column referring to the position of the piston.

Now, as to the manner in which the distribution of the steam is influenced by the motions above described, *first*, we must inquire, What are the duties which it is absolutely necessary that a slide valve should perform? These are really merely that it should admit steam to each end of the cylinder alternately, and should, either before or at the same time that it admits steam at the one end, provide egress for the steam which has already done its work at the other. In the case of the earlier locomotives the slide valves, instead of being proportioned as shown in Fig. 122, had no lap, or possessed it only to such slight extent as was necessary to insure that both steam ports should not be open for the admission of steam at one time. The action of this old form of valve will be first investigated. The positions of the valve corresponding to the commencement, to three intermediate points, and to the termination of the stroke of the piston, are represented by Figs. 1 to 5 (Fig. 125). In these diagrams the motion of the eccentric is, for the sake of convenience, communicated to the valve through a rocking shaft such as was frequently employed on old engines. This causes the position of the eccentric to be diametrically opposite to that which it would occupy if the motion was communicated from it directly to the valve. The angular advance is therefore *towards* the crank, instead of away from it, as would otherwise be the case, and it is just sufficient in amount to bring the edge of the face of the valve in a line with the outer edge of the steam port at the commencement of the stroke of the piston.

The proportions chosen for our example are as follows:—

Stroke of Piston, ...	20 inches.
Length of Connecting Rod, ...	5 feet.
Travel of Valve, ...	3 inches.
Outside Lap of Valve, ...	$\frac{1}{16}$ inch.
Inside Lap, ...	0 "
Outside Lead, ...	0 "
Inside Lead, ...	$\frac{1}{16}$ "
Width of Steam Ports, ...	1¼ inches.
Width of Exhaust Ports, ...	2½ "
Width of Bars, ...	1 inch.

In Fig. 1 (Fig. 125) the piston is shown at the commencement of its stroke, the slide valve having passed the centre of its stroke, by the amount due to the angular advance of its eccentric, which, as there is no outside lead, corresponds in this instance to a movement of the valve equal to the lap, or $\frac{1}{16}$ inch.

At the same time, the steam passage communicating with the end of the cylinder farthest from the piston is uncovered to the exhaust by the amount of the inside lead, which, as there is no inside lap, is also $\frac{1}{16}$ inch. In Fig. 2 the crank has performed one-eighth of a revolution, and both the steam ports are partially open, the one for the admission of steam, and the other for its egress. In Fig. 3 one-fourth of a revolution has been accomplished, and both ports are fully uncovered. In Fig. 4, the crank having made three-eighths of a revolution, the ports are again partially closed, the valve having assumed a position almost similar to that which it occupied in Fig. 2. In Fig. 5 the piston has reached the end of its stroke, and the steam port, which has hitherto been receiving steam, is uncovered $\frac{1}{16}$ inch on the exhaust side, the other steam port being entirely closed.

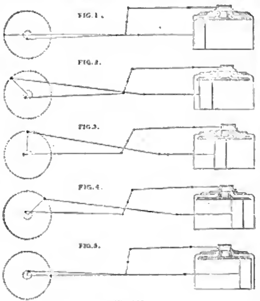

Fig. 125.

The form of diagram just described, although explaining pretty clearly the relative motions of valve and piston, does not show at a glance at what particular points in the stroke of the latter the admission, suppression, and exhaustion of the steam occur. For this purpose the diagram shown on Diagram Plate I is far better adapted. The proportions of the valve, &c., represented in this diagram are the same as in the last instance, the length of the eccentric rod being taken as 3 feet 6 inches.

The key to this diagram is, that in it the crank shaft is supposed to be fixed, and the whole of the engine is imagined to revolve round it. The method of constructing it is as follows:—Draw two straight lines intersecting each other at right angles, and consider the point of intersection to represent the centre of the crank shaft. Around this centre (A) describe a circle with a radius (A C) equal to the throw of the eccentric taken to any convenient scale; and upon one of the straight lines first drawn mark off the distance, A B, representing the length of the crank. This length may be laid off to the same scale as that employed for delineating the throw of the eccentric, or, if more convenient, any other scale may be used; but it must be remembered that whatever scale is adopted for the crank must also be used for the connecting rod, and for measuring the position of the piston in the manner to be afterwards described; whilst the scale employed when drawing in the throw of the eccentric, the length of the eccentric rod, and the proportions of the slide valve and ports, must also be the same throughout. In practice it is generally better to make the last mentioned scale the larger of the two, and that part of the diagram representing the motion of the piston can then be drawn within that relating to the motion of the valve. This is done in the case of the Diagram Plate I, the scale adopted for delineating the crank and its connections being half that employed for the eccentric, valve, &c.

Proceeding with the construction of the diagram, the next thing to be done is to mark off the position of the centre of the eccentric, and, as the engine is supposed to revolve instead of the crank shaft, the centre must be placed in the position it would occupy if the crank shaft revolved the reverse way, or *behind*, instead of in front of the crank. Thus, if it is intended that the diagram should be read in the direction of the arrow, the centre of the eccentric must be placed at C instead of at D, which is the point at which it would be situated if the crank shaft was supposed to rotate in the direction indicated.

There being no outside lead, the linear advance will be equal to the lap, or ⅞ inch, and, as the motion of the valve is derived directly from the eccentric, this will also equal the sine of the angular advance, or the distance corresponding to C D in Fig. 123. From the centre of the shaft describe two circles, one with a radius, A P, equal to the length of the connecting rod, *plus* that of the crank; and the other with a radius, A R, equal to the length of the connecting rod, *minus* that of the crank, the scale employed for these radii being the same as that to which the crank was drawn. These circles will represent the distance of the centre of the cross-head from the centre of the crank shaft at each end of the stroke. From the centre of the crank pin, with a radius equal to the length of the connecting rod, describe another circle which will touch the two just drawn at the two points corresponding with the terminations of the stroke of the piston. This circle is marked upon the diagram "path of piston," and if, through any point in it, for instance, Q, a radial line be drawn from the centre of the crank shaft, the distances, Q P, Q R, measured along this line, from such point, Q, to the circles marking the ends of the stroke, will equal the distances which the piston would be from the outer or crank ends of the stroke respectively, when the crank occupied a position bearing the same relation to the centre line of the engine that that of the radial line A P does to the horizontal centre line on the diagram.

Next, as to the motion of the valve. From the centre of the crank shaft describe another series of circles, representing the steam and exhaust ports and bars, as explained at the bottom of the diagram, the steam port communicating with the crank end of the cylinder being, of course, that nearest the centre. The radius of the circle denoting the centre line of the exhaust port is to be equal to the length of the eccentric rod, the scale used being that employed when laying out the position of the eccentric. In like manner describe around the centre of the eccentric a series of circles representing the paths of the edges of the slide-valve face, &c., the radius of the circle showing the path of the centre of the valve being equal to the length of the eccentric rod. Then, if the radial line A P be produced to L, the distance, L M, between the circles, representing the outside edge of the leading steam port and the outside edge of the corresponding valve face, is the amount to which the leading steam port is open for the admission of steam when the piston has travelled that part of its stroke represented by Q P, as explained above; and the space, N O, between the circles, representing the inside edge of the steam port at the crank end of the cylinder, and the corresponding inside edge of the slide-valve face, shows the extent to which that steam port is at the same time open to the exhaust. In the diagram the darkly-shaded spaces show the openings of the ports on the steam side, and those lightly shaded the openings to the exhaust.

The form of diagram just described may be considerably simplified by tracing out the motion of the centre line of the valve only, and obtaining the openings of the steam and exhaust ports in the manner which will be pointed out by the following considerations:—If a slide valve is without inside or outside lap, it is evident that the extent to which one port is opened for the admission of steam, by any movement of the valve from its middle position, is equal to that to which the other port is at the same time uncovered to the exhaust, and is also equal to the amount of such movement, or, in other words, to the distance between the centre line of the valve when so moved and the centre line of the exhaust port. If, however, the valve has either outside or inside lap, the openings of the ports on the steam and exhaust sides occasioned by this movement of the valve are less than the extent of such movement by the amounts of the outside or inside laps respectively. On the contrary, if the valve has inside clearance, the opening of the one port to the exhaust is greater than the distance moved by the valve by the amount of such clearance. We therefore see that in a diagram similar to the one just explained one circle showing the path of the centre of the valve may be substituted for the four showing the motions of each edge of the face; and in place of six circles being described around the centre of the crank shaft, to represent the edges of the steam and exhaust ports, three may be employed—one described with the length of the eccentric rod as a radius, and two others, one within and one without this circle, at a distance from it equal to the outside lap of the valve. If the valve has inside lap, two other circles will be necessary, similarly placed to those just mentioned, except that their distance from the first named circle must equal the inside lap. These circles will serve as datum lines from which the opening of the ports for the admission and egress of steam may be measured.

Fig. 126 shows to an enlarged scale a portion of a diagram similar to that on Diagram Plate I., and on the left side of it the same portion simplified in the manner just explained. In this part of the figure the portion of a circle

A A represents the path of the centre line of the valve; that marked B B being the one described from the centre of the crank shaft, with the length of the eccentric rod as a radius, and representing the centre line of the exhaust port. The parts of circles C C, D D, on each side of it, are distant from it by the amount of the outside lap of the valve, which, to make the diagram clearer, has in this example been increased to ⅛ inch, whilst the other proportions remain the same as those employed upon Diagram Plate I.

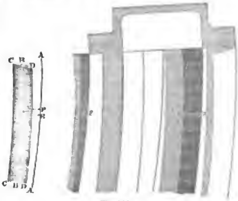

Fig. 126.

The distances which represent the openings of the ports to the steam and exhaust are correspondingly lettered in the two parts of the figure, and it will thus be seen that, as the valve has no inside lap, the distance E E, measured radially from the circle of the centre line of valve to the circle of the centre of the exhaust port, is equal to the opening of one of the ports on the exhaust side; whilst the distance F F, similarly measured, by which the circle of centre of valve is outside the outer, or inside the inner circle of lap, is equal to the amount to which the other port is uncovered on the steam side. The fact of the circle of centre of valve being in the position shown in the figure—namely, outside that of the centre of the exhaust—of course indicates that the port which is open for the admission of steam is the one nearest the crank shaft. When the circle showing the path of the centre of valve falls between the circles of lap, both ports are closed to steam.

The form of diagram we have just described, and which we believe was first proposed by Mr. J. F. Gray, is, when drawn to a large scale, probably the best that can be adopted for showing the relative motions of the piston and slide valve, if the latter is driven by a single eccentric, as the influence of the varying obliquity of the connecting and eccentric rods is clearly shown for every point in the revolution of the crank shaft.

As none but circular curves are employed, it can also be laid down very quickly. In a work like the present, however, it necessarily happens that the size of the plates is limited; and as in this kind of diagram it is difficult, when the scale is small, to determine the precise points at which the steam is cut off, &c., we shall, when treating of the effects of variation in the proportions of the slide valve, describe some other forms more suitable to the space at our disposal.

Let us now consider what information we can gain from the Diagram Plate I. respecting the distribution of the steam effected by the valve there represented. Beginning with the commencement of the stroke towards the crank shaft, we find that the leading steam port is just on the point of opening on the steam side, and that the other, communicating with the crank end of the cylinder, is open to the exhaust by the amount of the lap, or $\frac{1}{16}$ inch. We also see that, as the centre line of the eccentric is at this time almost at right angles to the centre line of the engine, the valve is at this part of the revolution moving very rapidly; and, tracing out its motion, we find that the one port becomes fully open to the exhaust when the point S is reached, the piston having by that time performed about 5¼ inches of its stroke.

The complete opening of the other port for the admission of steam occurs a little later at T, the piston having moved about 5¾ inches. From this point both ports remain entirely uncovered, the one for the admission of the steam, and the other for its exhaustion, until the point U is arrived at, when that which is open to the exhaust begins to close, the piston having traversed about 15¼ inches. A little further on, at V, the port open to steam also begins to close, the piston having then performed nearly 16 inches of its stroke. The only important points which remain to be noticed are, the closing at F of the port which has been receiving steam, and the opening of the same port to the exhaust, which occurs at G simultaneously with the entire closure of the port which has hitherto been exhausting. Although both these events take place before the completion of the half revolution of the crank shaft, yet it is evident from the diagram that the piston may for all practical purposes be considered to have arrived at the end of its stroke before their occurrence.

It needs but a slight consideration of the movements of the valve just described to assure us that its action is exceedingly defective. The fact of the port which should receive the steam being entirely closed at the beginning of the stroke, whilst the other is only very slightly open to the exhaust, will evidently cause an excessive back pressure to be exerted in direct opposition to the motion of the piston; and the first part of the stroke must therefore be performed by the aid of the other cylinder (in the case of a double engine), or by the momentum stored in the rotating parts during the previous portion of the revolution. Moreover, as the steam is admitted until nearly the end of the stroke, a violent strain is placed upon the machinery by the necessity which exists for arresting the motion of the piston and other reciprocating parts, entirely by the aid of the connecting rod and crank, instead of their action being assisted by the pre-admission of steam to that end of the cylinder which the piston is approaching. The diagram shows that we are indebted to the $\frac{1}{16}$-inch lap for the slight opening of the one port to the exhaust before the end of the stroke; and this important property of lap will be more apparent when we analyze the action of a valve of more modern proportions. This we shall now proceed to do, and, as mentioned above, shall embrace the opportunity thus afforded to describe another kind of diagram. The dimensions chosen for the valve, &c., are as follow:—

Stroke of Piston, ...	2 feet.
Length of Connecting Rod, ...	6 „
Travel of Valve, ...	5¼ inches.
Outside Lap of Valve, ...	1⅜ „
Inside Lap of Valve, ...	0 inch.
Outside Lead of Valve, ...	0⅛ „
Inside Lead of Valve, ...	1½ inches.

Width of Ports, &c.—Steam, 1¼ ins.; Exhaust, 3 ins.; Bars, 1 in.

The form of diagram to which we have just alluded is represented on Diagram Plate II., Fig. 1. Its construction is as follows:—Draw a straight line, A B, and from any point, C, in it, as a centre, describe two circles, one with a radius, C A, equal to the length of the crank, and the other with the radius, C D, equal to the throw of the eccentric, any convenient scale being employed, the larger the better. Through the points A and B, where the larger circle intersects the line first drawn, draw lines E E, F F at right angles to A B, and, upon one of these, as F F, mark off the widths of the steam and exhaust ports, making the point B the centre of the latter. From the points marking the edges of these ports draw lines parallel to A B, and extending to E E. Next let it be imagined that the reciprocations of the piston and valve, instead of being parallel, occur at right angles to each other—those of the piston taking place in the direction of the line A B; whilst those of the valve are performed at right angles to that line. Divide both the circles into any similar number of convenient parts (in the figure sixteen is the number of divisions adopted), these divisions commencing, in the case of the circle representing the path of the crank pin, at the point B, and, in the case of the smaller circle, at a point G, distant from the line A B by the amount H G, which is equal to the lap of the valve added to the lead.

The points of division of both circles should be correspondingly numbered, commencing at the points B and G respectively; and any point on the circle representing the path of the centre of the eccentric will then show the position of that centre, agreeing with the similarly numbered position of the crank pin.

It will be noticed that, on account of the motion of the valve being supposed to take place at right angles to that of the piston, the eccentric is situated one-fourth of a revolution in arrear of the position which it would occupy on an engine.

We have explained in an early part of this chapter, that when the alterations in the motions of the piston and valve, caused by the varying obliquity of the connecting and eccentric rods, are disregarded, the position of the piston and valve corresponding to any point in the revolution of the crank shaft may be determined by means of ordinates drawn, as in Fig. 124, from the centres of the crank pin or eccentric respectively. Let the action of the connecting and eccentric rods be neglected in the present instance, and the positions of the piston corresponding to the occupation by the crank pin of the points 1, 2, &c., can then be obtained by ordinates drawn from these points at right angles to A B. Similarly the positions of the centre line of the slide valve will be represented by lines drawn from the points 1, 2, &c., on the inner circle, *parallel* to A B. If a curve be now drawn through the points where the similarly numbered ordinates of each set intersect each other, then the distance of this curve (measured parallel to E E) from the line A B at any point is equal to the distance which the valve is removed from the centre of its travel when the piston has arrived at the point referred to. For example, if, from the point J, in the line A B, we draw the perpendicular J K, reaching to the curve showing the motion of the valve, then the distance, J K, is equal to the amount by which the valve is removed from the centre of its travel when the piston has traversed the portion of its stroke represented by B J.

This elliptical curve shows, as has just been mentioned, the position of the centre of the valve. To represent the motion of the edges of its face, it is only necessary to draw curves similar to that already existing, and removed from it by spaces equal to the distances of such edges from the centre of the valve, the spaces between the curves being measured at right angles to A B. In the figure the positions assumed by the outer edge of one face and the inner edge of the other, during the stroke of the piston from B to A, are delineated in this manner, the one set being represented by the line *a b c d*, and the other by that marked *e f g h*.

Both these curves are parallel to that first referred to, and are distant from it by the spaces 1′ *a* and *e* 1′ respectively. The darkly shaded portion enclosed between the curve *a b c d* and the line representing the outer edge of the steam port shows the extent to which this port is uncovered on the steam side; whilst the more lightly shaded area, bounded by the inner edge of the other port on the one side, and partly by the outer edge of the same port, and partly by the curve *e f g h*, on the other, represents the amount to which this port is open for eduction. The very darkly shaded portion at *o* shows the opening of the port for the pre-admission of steam before the end of the stroke.

When treating of the diagram on Diagram Plate I, we explained how the points of admission, suppression, &c., could be determined by means of a line showing the motion of the centre of the valve only; and the same principles are of course applicable to the simplification of the diagram now before us.

Fig. 2, Diagram Plate II., represents the diagram thus simplified. In this case the curve showing the motion of the centre of the valve is obtained in precisely the same way as that in Fig. 1 on the same Plate, and the amount to which the one port is, at any point in the stroke of the piston, open to the exhaust is determined by measuring the distance of the curve at that point from the centre line A B; whilst the distance to which the curve projects beyond the lines C C or D D, as the case may be, represents the extent to which the other port is uncovered for the admission of steam, the two lines, C C, D D, being drawn parallel to A B, and separated from it by a distance equal to the lap of the valve. Thus, when the piston is at the point L, the one port is open to steam to the extent M N, and the other is uncovered to the exhaust to the amount N L. If the distance last mentioned exceeds the width of the steam port, it signifies that the inside edge of the valve face has travelled past the port, which is of course fully open. If the valve had inside lap or clearance, it would be necessary to add two more lines to the diagram. These would be drawn one on each side of A B, parallel to it, and at a distance from it equal to the amount of such inside lap or clearance, and would then form the datum lines from which the opening of the ports to the exhaust would be measured. If the valve had inside lap, the distance of the curve from the nearest of the two lines would equal the opening of the port for eduction; but if it had inside clearance, the furthest of the lines would have to be employed as the datum from which the amount of this opening must be measured.

Let us now compare the distribution of the steam effected by the valve delineated on this diagram with that of the old valve shown on Diagram Plate I. In the first place, we find that at the commencement of the stroke the one steam port is uncovered by the amount of the lead, or ¼ inch, and, what is even more important, that the other port is fully open for eduction. Following up the course of the stroke, we see that, owing to the greater angular advance of the eccentric, the full opening of the port which is receiving steam is sooner arrived at than was the case when the valve without lap was employed. In the example this full opening occurs at *b*, when the piston has accomplished about 5½ inches, or 23 per cent. of a 24-inch stroke. In the case of the old valve it will be remembered that the port was not fully uncovered until about 5¾ inches, or 28¾ per cent. of the 20-inch stroke of the piston, had been performed. Proceeding further, we find that the steam, instead of being admitted until nearly the end of the stroke, is suppressed at *c*, when the piston has accomplished about 75 per cent. of its travel. From the point *c* expansion is carried on until the piston arrives at that marked *i*, when the port which has been receiving steam begins to open on the exhaust side, the curve *i r* showing the motion of the inner edge of the valve face. At the same time the eduction from the other port ceases, and the steam, which has not yet escaped from that end of the cylinder with which it is connected, is subjected to a gradual compression during the remainder of the stroke, thus forming a cushion by which the motion of the piston and other reciprocating parts is arrested. Shortly before the end of the stroke the port now under consideration is opened for the admission of steam, as shown at *o*, and, when the termination of the stroke is arrived at, each of the ports has assumed the condition possessed by the other at its commencement.

The distribution of steam above described is *apparently* much superior to that effected by the old valve; but we must here remark that neither the foregoing nor any other form of geometrical diagram can give us any *absolute* proof of the advantages gained by the use of any particular proportions of lap, lead, or travel. It is true that, reasoning upon the facts made known to us by the assistance of such diagrams, we may form an opinion (and probably a correct one) of the adaptability of any particular valve gear to the purpose for which it is employed; but at the same time conclusive evidence regarding the perfection or imperfection of such valve gear can be obtained from indicator diagrams alone.

In the remarks upon the distribution of steam effected by the valve last described, no notice was taken of the inequalities produced by the action of the connecting rod. In the case of the old valve the point of cut-off was so near the end of the stroke that this action was inappreciable; but with a valve having the proportions adopted in Diagram Plate II., however, the circumstances are very different, as the suppression occurs at a point in the revolution of the crank shaft where the obliquity of the connecting rod exercises considerable influence upon the position of the piston. To ascertain the amount of irregularity arising from this cause we may proceed as follows:—In place of employing straight ordinates, drawn from the points 2, 3, &c., in the circle of the path of the crank pin, to represent the position of the piston, join those points similarly situated on each side of the centre line (such as 2 and 16, 3 and 15, &c.) by circular arcs, the radius of which is equal to the length of the connecting rod (in the example, 6 feet), taken to the same scale as that employed for drawing the circle of the path of the crank.*

The effect of the obliquity of the connecting rod, as explained by Fig. 124, being always to draw the piston nearer to the crank shaft than it would be if the rod was of infinite length, the *convex* side of the arc must be towards that end of the line A B which is supposed to represent the crank-shaft end of the stroke. Similar arcs must also be drawn from the points *k* and *l*, the former of which represents the position of the crank pin when the suppression of the steam occurs, and the latter its situation when the exhaustion commences. The position of the piston corresponding to these two points will then be indicated by the intersection of the arcs with the centre line, A B, at the points *m* and *n* respectively; and it will thus be seen that when the stroke is being made from B to A, the period of admission is prolonged by the amount *m o*, which is equal to about ¾ inch, or 3⅛ per cent. of the stroke. The release and compression are, under the same circumstances, deferred to the extent of the space *n p*, equal to $\frac{5}{16}$ths of an inch, or about 1¼ per cent. When, on the contrary, the stroke is performed from A to B, the admission is curtailed, and the release hastened by the per centages just mentioned.

With a connecting rod three times the length of the stroke, and a valve of the proportions under consideration, there will therefore be a difference of 6¼ per cent. between

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* The reason why these arcs give the corrected position due to the action of the connecting rod will be readily explained by a consideration of the following diagram, Fig. 127.

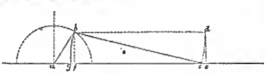

Fig. 127.

Let *a b* represent the crank, and *b c* the connecting rod attached to it; then, from the point *b*, with the radius *b c*, describe the arc *c d*; and from the point *d*, where this arc cuts the line *b d* (drawn through *b* parallel to *a c*), let fall the line *d e* perpendicular to *a e*. Also, from the point *b* let fall the perpendicular *b f*, and from the centre *c*, with the radius *c b*, describe the arc *b g*, which will correspond with the arcs drawn on Fig. 1, Diagram Plate II., above alluded to, except that the direction of the convexity is reversed. It will then be evident that if it was possible to maintain the connecting rod parallel to the centre line of the engine, during the revolution of the crank, the centre of the cross-head would be situated at *e*, when the crank was in the position shown in the diagram. As, however, this is not the case, the obliquity of the connecting rod causes an alteration in the position of the cross-head equal to *c e*, which, as the arcs *c d* and *b g* are precisely similar, is also equal *g f*.

the periods of admission, and 2½ per cent. between the position of the points of release, during the two strokes of the piston towards and away from the crank shaft, and the differences will evidently be augmented by any such alteration in the proportion of the valve, ports, &c., as may cause the suppression or release to take place nearer the centre of the stroke, and consequently at a time when the obliquity of the connecting rod would be greater. As regards the point of pre-admission, it is, with all ordinary proportions of lead, so near the end of the stroke that the action of the connecting rod upon it is practically imperceptible.

We have dwelt at some length upon the diagram last described, as it is one which has been very generally employed to illustrate the motion of the slide valve—a matter which it certainly explains very clearly. It has, however, some disadvantages—one of these being that unless the divisional points of the circles from which the elliptical curve is projected are very numerous, no great accuracy can be obtained. This remark particularly applies to those parts of the curve representing the motion of the valve near the commencement and termination of the stroke of the piston. An increase in the number of divisions of course entails an increased and sometimes objectionable expenditure of time in the delineation of the diagram; and it will therefore be advisable that we should, before leaving this portion of our subject, describe another form of diagram, by the use of which all the desired information can be obtained, not only much more rapidly, but also with perfect accuracy.

This diagram is shown by Fig. 128, and is constructed as follows:—Draw a straight line, A C, of a length equal to the travel of the valve; bisect it, and from the point of bisection, B, as a centre describe a circle with a radius equal to B A. Next draw the line D E parallel to A C, and distant from it by the sum of the lap and lead, or linear advance of the valve; and from the point where this line cuts the circle draw the line E O, passing through the centre, B. Parallel to the line E O, and on each side of it, draw the lines P P′, Q Q′, each distant from it by the amount of the lap, F G. If the valve has inside lap or clearance, draw the lines U U′, V V′, also parallel to E O, and separated from it by a distance equal to such lap or clearance. Then draw the parallelogram H I K L (the lines H K and I L being at right angles to A C), and suppose H I or K L to represent the stroke of the piston.

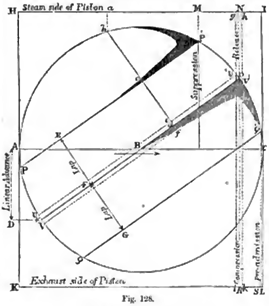

Fig. 128.

The manner in which the opening of the ports corresponding to any point in the stroke of the piston may be ascertained from this diagram will be best explained by an example. Let it be required, for instance, to determine the extent to which the ports are open for the admission and exhaustion of steam when the piston has traversed that portion of its stroke represented by H *a*. Draw *a b* parallel to A H, and from the point where this line cuts the circle, draw *b d* perpendicular to E O. Then the extent to which one port is open for the admission of steam is equal to the distance *b c*; whilst the other is, if the valve has no inside lap or clearance, uncovered to the exhaust by the amount *b d*.

If the valve has inside lap or clearance, let their amount be represented by *d e* or *d f* respectively, and the opening of the port to the exhaust will then, in the former case, be diminished to *b e*, and in the latter increased to *b f*. If the opening for eduction thus indicated exceeds the width of the steam port, it shows, as explained in the description of the last diagram, that the port is full open, and that the inner edge of the valve face has travelled past it. From the example just given it will be seen that the lines P P′, E O, and Q Q′ bear precisely the same relation to the circle that the steam and exhaust datum lines did to the elliptical curve in the diagram on Diagram Plate II., the opening of the ports for admission or eduction being determined in the same manner; the motion of the piston, however, being measured in the direction A C, instead of along the datum lines.

If, as is done in Fig. 128, we suppose those events which relate to the steam side of the piston to be referred to the line H I, while those relating to the exhaust side are referred to the line K L, we find that suppression occurs at M, that point being projected from that marked P where the line P P′ cuts the circle. In the same way (if the valve has no inside lap or clearance) we obtain the position of the points of release, N, and compression, R, by projection from the point O, and that of pre-admission, S, from the point Q. If the valve has inside lap equal to *d e* or *e f*, compression will commence at *i*, and release be deferred until the point *h* is reached; if, on the contrary, it is provided with inside clearance to a similar extent, release will take place at *g*, and the compression will be postponed until the point *k* is arrived at by the piston. In this diagram it will be seen that the length of the stroke of the piston is represented by the travel of the valve. As, however, this frequently happens to be an awkward scale, it is in most cases better to employ the slight modification of this diagram shown by Fig. 129. In this form of the diagram the path of the crank is denoted by a separate circle, the diameter, T T, of which represents to any convenient scale the stroke of the piston. The positions of the crank corresponding to the points of suppression, &c., are determined by radial lines drawn through the points O, P′, Q′, &c., on the circle representing the path of the centre of the eccentric; and the place occupied by the piston may, of course, be then ascertained by projection from the points where these radial lines cut the circle of the path of the crank pin. The construction of the rest of the diagram is the same as that last described.

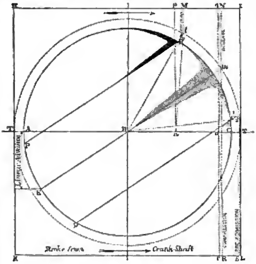

Fig. 129.

We must next explain how the irregularities due to the action of the connecting rod can be shown on this form of diagram. To do this it is merely necessary to substitute curved for straight ordinates, as shown by the dotted lines *l n* and *m o* on Fig. 129. These lines are to be drawn with a radius equal to the length of the connecting rod, taken to the same scale as that to which the circle representing the path of the crank pin has been already laid down, and the position of the piston corresponding to the occurrence of the suppression, &c., of the steam can then be obtained by projecting from the points where the curved lines intersect the line T T. The centre from which the curves are described must be situated upon the line T T produced, and the curves themselves must, as in the diagram on Diagram Plate II., be drawn so that the convex side of them is towards that end of the line T T which is supposed to be nearest the crank shaft. The curved lines on the figure show that during the stroke of the piston from left to right, the suppression and release of the steam are accelerated by the amounts *p* M and *q* N respectively, through the action of the connecting rod.

To counteract the inequality in the periods of admission caused by the action of the connecting rod, the slide valve has been sometimes made with unequal lap, that on the edge of the valve farthest from the crank shaft being of course the greatest. The same effect as that produced by unequal lap may also be obtained by making one of the steam ports wider than the other; and this plan is preferable to that of using unequal lap, as no harm can ensue if the valve is accidentally placed in the wrong way. Both plans have, however, the defect of giving unequal lead. The manner in which the difference of lap requisite to neutralize the action of any given connecting rod can be determined, will be explained hereafter.

We have now described in general terms the action of a slide valve and single eccentric, and the manner in which their action may be investigated geometrically; and in the following chapter we shall proceed to consider more definitely the effects produced by variations in their proportions.

CHAPTER VII.[*]

ON THE PROPORTIONS OF VALVES AND ECCENTRICS.

Before treating independently the effect of variations in the lap, lead, and travel of a slide valve, it is advisable that it should be clearly understood that a difference in the distribution of the steam will be produced only by an alteration of the *relative* proportions which these elements bear to each other. Thus, whether the lap, lead, or travel of a slide valve are *all* doubled or *all* halved, the per centages of admission, expansion, &c., will remain the same (the size of the ports being supposed to be increased or diminished in the same proportion); and this will be clearly understood if it is considered that the valve may in the latter case be supposed to be merely made to one-fourth the scale employed in the former instance. If, however, either the lap, lead, or travel be altered by itself relatively to the other two elements, the distribution of the steam is at once modified, and the extent of such variation produced by any given alteration in the proportions of a valve and eccentric shall be investigated forthwith.

The form of diagram which shall be employed is a slight modification of that described in the last chapter. The periods of admission, &c., shall be expressed in per centages of the stroke of the piston; and as the object in view is an analysis of the action of valves of different proportions, the disturbances produced by the connecting rod shall be disregarded.

In order that we may determine the relative advantages of various proportions of lap, lead, and travel, it is necessary that we should at the outset be thoroughly acquainted with the properties which it is desirable that a valve gear should possess when employed to work steam expansively. These may be briefly stated as follows:—

1st. That it should admit steam freely to the cylinder during the whole of that portion of the stroke when such admission is required, so that no loss of pressure may ensue from the ports being insufficiently uncovered.

2d. That it should effect the cut-off of the steam as suddenly as possible, in order that "wire-drawing" or gradual reduction of pressure may be avoided.

3d. That it should afford as long a period for expansion as is consistent with the attainment of the fourth qualification.

4th. That it should cause the release of the steam from the one side of the piston, and the compression of that remaining in the cylinder on the other, to commence at such a point in the stroke as in the one case to avoid unnecessary back pressure during the following stroke, and in the other to provide sufficient resistance to absorb the momentum acquired by the reciprocating parts of the engine.

5th. That it should give an amount of pre-admission just sufficient to enable the steam to acquire its full pressure in the cylinder either by the commencement or almost immediately after the commencement of the stroke of the piston.

And, 6th. That it should fulfil the above requirements symmetrically for both strokes of the piston.

We may now proceed to analyze the action of some of the proportions of slide valves and eccentrics in general use, and to explain the manner in which alterations in the relative proportions of the elements, lap, lead, and travel, affect the distribution of the steam. When the defects of the old valve, shown on Diagram Plate I., were discovered, it was at first attempted to remedy them by an increased amount of lead; then followed the addition of lap; and lastly, varying travel was introduced as a means of regulating the extent of expansion. In the present chapter the variations of the different proportions shall be considered in the following order:—1st. Travel and lap constant, lead varying; 2d. Travel and lead constant, lap varying; 3d. Lap and lead constant, travel varying; 4th. Lap constant, lead and travel varying simultaneously; 5th. Lead constant, travel and lap both varying.

1st. *Travel and lap constant, lead varying.*—Fig. 130

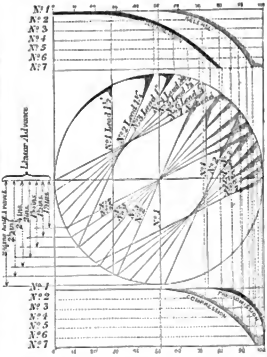

Fig. 130.—Diagram showing the Effect of Varying Lead.

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[*] By Mr. W. H. Maw, *vide* foot-note, p. 139.

shows the action of a slide valve having 5½ inches travel and 1¼ inches lap, the lead varying by equal decrements of ¼ inch from 1½ inches to *nil*. The points of cut-off and release corresponding to each proportion of lead are projected upon lines at the top of the diagram representing the stroke of the piston, whilst those at which the compression commences, and pre-admission occurs, are in a similar manner referred to the bottom of the figure; and curved lines drawn through each set of points thus projected show very clearly by the variation in their curvature the effect of the equal diminutions of lead. The most noticeable defects of lead, when employed as a means of regulating the amount of expansion, are—*firstly*, the comparatively limited extent to which such expansion can be carried by its agency; *secondly*, the short portion of the stroke during which the expansion so produced is allowed to continue before the release of the steam occurs; and, *thirdly*, the large amount of pre-admission given. An investigation of the diagram will show that the shortest portion of the stroke which can take place before the steam is cut off must bear the same proportion to the whole stroke of the piston that the lead of the valve does to its whole travel, in the instance before us, 27·27 per cent.; and it will also be apparent that even this moderately early suppression can only be obtained by giving the eccentric an angular advance of 90°, and consequently (unless the valve has inside lap) permitting the release and compression to commence at half-stroke, the expansion continuing during 22·73 per cent. of the stroke only. The pre-admission is also in this case very excessive, being equal in amount to the admission, or 27·27 per cent. of the stroke; and, in fact, this proportion of lead is one which is perfectly inadmissible in practice. It should be noticed, however, that, with varying lead, the full opening is given to the ports at all grades of expansion produced by it, and this is an advantage not possessed by either varying lap or varying travel. Notwithstanding this, it is evident that varying lead *alone* is quite unsuited for regulating the amount of expansion; but when occurring in conjunction with varying travel, it has some points in its favour, which will be referred to hereafter, when treating of the simultaneous variation of these two elements.

2*d. Lead and travel constant, lap varying.*—The effect upon the distribution of the steam produced by varying the lap of a valve is shown by Fig. 131, on which is delineated the action of a valve having 5½ inches travel, ¼ inch lead, and lap decreasing from 2½ inches to ½ inch. The points of suppression and release are, as in the diagram showing the effect of varying lead, referred to the upper, and those of compression and pre-admission to the lower part of the figure; and it will be at once seen that the use of varying lap gives the power of producing a far greater degree of expansion than was attainable by the employment of varying lead. The shortest period of admission which can be obtained bears, as in the last case, the same proportion to the whole stroke of the piston that the lead of the valve does to the whole travel; but as in the present instance this lead does not require to be increased with the earlier cut-off, the period of admission may be correspondingly less. With the proportion of lead and travel represented in the diagram, the limit is reached when the lap amounts to 2½ inches, the per centage of admission being in that case only 4·55 per cent. of the stroke. Even with this early cut-off the release does not occur until half-stroke, thus allowing the steam to expand during 44·45 per cent. of the stroke, whilst the pre-admission only amounts to 4·55 per cent.; the extreme opening of the port for the admission of steam is, however, in this case, reduced to ¼ inch only.

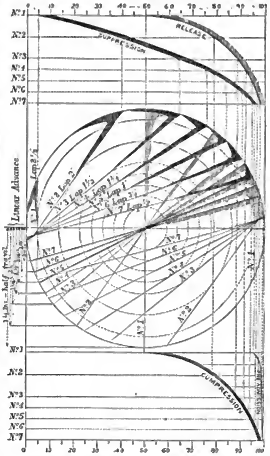

Fig. 131.—Diagram showing the Effect of Varying Lap.

3*d. Lap and lead constant, travel varying.*—The influence of varying travel upon the points of suppression, &c., is represented upon Fig. 132, the valve shown on which has 1¼ inches lap, ¼ inch lead, and travel varying from 6 inches to 2½ inches; and as this diagram differs slightly in its arrangement from those previously employed, a few words of explanation respecting it may probably be necessary. It is evident that in those diagrams in which the stroke of the piston is represented by the travel of the valve, the introduction of varying travel would of course necessitate the employment of a series of different scales for measuring the motion of the piston, unless some plan was adopted for reducing them to one standard. One way in which this can be done was explained by Fig. 129, and another is shown by the diagram now under consideration. According to the plan adopted in the latter case, the same steam and exhaust datum lines are employed for all the different proportions of travel; and the centres of all the circles representing these various travels are so situated upon the exhaust datum line, A A, that all the circles pass through the point, B, from which the angular advance is measured. The points of suppression, release, &c., corresponding to any particular travel are marked off upon a diameter of the circle representing such travel, drawn through the point B, and are then projected, by means of lines parallel to the datum line A A, upon the diameter B C, which is chosen to represent the stroke of the piston. From this line they are referred to the top and bottom of the figure, as in the last two instances.

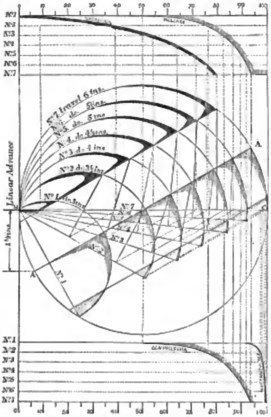

Fig. 132.—Diagram showing the Effect of Varying Travel.

Now, as regards the action of varying travel,—Like varying lap, it gives a power of producing expansion far superior to that attainable by the use of varying lead; and as was the case with both varying lead and lap, the limits of expansion are reached when the angular advance of the eccentric becomes 90°, the travel being at the same time equal to twice the lap and lead, and the distance travelled by the piston before the steam is cut off forming, as in the previous instance, the same fraction of its whole stroke that the lead of the valve does of its entire travel; or, with the proportions shown on the figure, 8·33 per cent. As the angular advance is 90°, the pre-admission is equal to the admission, and the valve, unless provided with inside lap, releases the steam at half-stroke, the extreme opening of the port being, as in the case of varying lap, reduced to the amount of the lead, or ¼ inch.

Before proceeding with the fourth and fifth division of this chapter relating to the simultaneous variation of two of the elements, it is advisable that we should summarize the results obtained from the three preceding diagrams. This is done in the following Table, and a consideration of it, in conjunction with the diagrams to which it refers, will enable us to determine, approximately, to what extent a slide valve driven by a single eccentric fulfils the conditions mentioned as desirable at the commencement of this chapter, and also to form an opinion as to the most eligible of the three modes of regulating the amount of expansion effected by it.

TABLE SHOWING THE EFFECT OF VARIATIONS IN THE PROPORTIONS OF THE LEAD, LAP, AND TRAVEL OF A SLIDE VALVE UPON THE DISTRIBUTION OF THE STEAM.

Lead.		Fig. 130.	Lead Varying. Lap, 1¼ Ins. Travel, 5½ Ins.			
				Positions (expressed in per centages of the stroke of the piston) of the points of		
		Linear Advance.	Angular Advance.	Suppression.	Compression and Release.	Pre-admission.
No. 1	1½ ins.	2¾ ins.	90° 0′	27·27	50·0	72·73
2	1¼	2½	65 20	47·96	70·86	89·22
3	1	2¼	54 50	57·1	78·79	94·21
4	¾	2	46 40	64·06	84·31	97·06
5	½	1¾	39 30	69·85	88·58	98·51
6	¼	1½	33 0	75·0	91·93	99·72
7	0	1¼	27 0	79·4	94·55	100·0
Lap.		Fig. 131.	Lap Varying. Travel, 5½ Ins. Lead, ¼ In.			
No. 1	2½ ins.	2¾ ins.	90° 0′	4·55	50·0	95·45
2	2	2¼	54 50	40·04	78·79	99·49
3	1½	1¾	39 30	65·03	88·58	99·67
4	1¼	1½	33 0	75·0	91·93	99·72
5	1	1¼	27 0	83·23	94·55	99·75
6	¾	1	21 20	89·84	96·52	99·77
7	½	¾	15 50	94·81	98·1	99·78
Travel.		Fig. 132.	Travel Varying. Lap, 1¼ Ins. Lead, ¼ In.			
No. 1	3 ins.	1½ ins.	90° 0′	8·33	50·0	91·67
2	3½	1½	58 50	37·49	75·87	98·69
3	4	1½	48 40	52·32	83·02	99·24
4	4½	1½	41 50	62·52	87·25	99·49
5	5	1½	37 0	69·53	89·93	99·62
6	5½	1½	33 0	75·0	91·93	99·72
7	6	1½	30 0	78·91	93·3	99·78

As regards the first qualification,—free admission of the steam up to the point of cut-off,—it will be seen that the best results are attained when lead is the varying element—the extent to which the ports are uncovered for the admission of steam being in that case the same with all degrees of expansion. In the case of both varying lap and varying travel the distance which the ports are opened decreases as the cut-off becomes earlier; it being, of course, always equal to the half-travel, *minus* the lap. The maintenance of a sufficient supply of steam to the cylinder, up to the point of suppression, is of course much influenced by the suddenness with which the cut-off is effected; and this leads to the second qualification. The suppression of the steam would occur most suddenly if effected when the valve was at half-stroke (at which time it is moving most rapidly); and it is clear, that with any given travel, the more nearly the valve approximates to this position at the time the cut-off is effected, the more sudden this cut-off will be. As, moreover, the distance of the valve from the centre of its travel, at the instant when it closes the steam-port, depends upon the proportion which the lap bears to the travel, it is evident that either an increase of the former or a diminution of the latter will cause the closing to take place more gradually, and, as a necessary consequence, tend to produce "wire-drawing" of the steam. When lead only is the varying element, the proportions of lap and travel remaining constant during all grades of expansion, the speed with which the valve is moving at the time of suppression is the same at whatever point in the stroke of the piston this cut-off is effected; and in this respect, therefore, varying lead possesses an advantage over the other two modes of regulating the amount of expansion, as in both of them the velocity of the valve at the time of cut-off decreases as this occurs earlier in the stroke. The relative position held by varying travel and varying lap, with respect to suddenness of cut-off, we shall be enabled to estimate more clearly when we arrive at the fifth division of this chapter, treating of their simultaneous variation.

The third qualification mentioned as desirable was, that the steam, after being cut off, should be allowed to expand during as long a portion of the stroke of the piston as was compatible with a free release. The Table above given at once shows that this requirement is fulfilled in very different degrees by the three modes of varying the expansion now under consideration, the action of varying lead being, as before-mentioned, particularly faulty in this respect. For instance, if we compare the examples No. 2 on the three diagrams (Figs. 130, 131, and 132), we find by that showing varying lead, that the steam, when cut off at 47·96 per cent. of the stroke of the piston, is released when 70·86 per cent. has been accomplished, and is thus allowed to expand to only $\frac{70·86}{47·96} = 1·47$ times its original bulk. In the case of No. 2, on the diagram showing the influence of varying lap, the cut-off is effected at 40·04 per cent. of the stroke, or 7·92 per cent. earlier than in the example just mentioned; yet, notwithstanding this, the release does not occur until the piston has travelled 78·79 per cent. of its stroke, the expansion continuing through 38·75 per cent., and the steam increasing its volume $\frac{78·79}{40·04} = 1·96$ times. With varying travel the case is very similar, the cut-off and release occurring at 37·49 and 75·87 per cent. of the stroke respectively, and the steam attaining $\frac{75·87}{37·49} = 2·02$ times its initial volume before being permitted to escape.

In order to show the relative periods of expansion afforded by the whole of the various proportions mentioned in the table, we may with advantage employ the form of diagram* represented by Fig. 133. This diagram (which represents the expansion curve for the various proportions of lead enumerated in the first division of the Table) is constructed as follows:—Draw the line A B, representing to any convenient scale the stroke of the piston. From the point A lay off distances corresponding to the periods of admission of each of the examples 1 to 7 in the Table; and then at each of the points so marked erect a perpendicular, representing by its height the period during which the expansion of the steam is allowed to continue in the case of the particular example to which it refers. In Fig 133 these perpendiculars are drawn to the same scale as the line A B, and their lengths are as follow:—

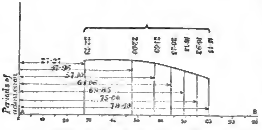

Fig. 133.—Diagram showing the Expansion Curve for Varying Lead.

No. 1 = 50·0 — 27·27 = 22·73
" 2 = 70·86 — 47·96 = 22·9
" 3 = 78·79 — 57·1 = 21·69
" 4 = 84·31 — 64·06 = 20·25
No. 5 = 88·58 — 69·85 = 18·73
" 6 = 91·93 — 75· = 16·93
" 7 = 94·55 — 79·4 = 15·15

Through the upper ends of the ordinates is drawn a curve which will show by the length of ordinates drawn to it the period of expansion corresponding to any cut-off between examples 1 and 7. In Fig 134 the curve just described is repeated, and is so placed that it can be compared with the two other curves on the same figure, representing the periods of expansion allowed by the various proportions of lap and travel mentioned in the Table.

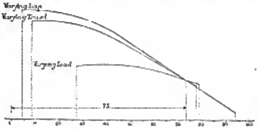

Fig. 134.—Expansion Curves for Varying Lap, Travel, and Lead.

It must be remembered that although these curves are intended to show the effects of varying lap, travel, and lead generally, the precise results indicated by them are only those due to these variations when acting upon valves having the proportions mentioned in the foregoing Table. It will be noticed that all the curves intersect at the point corresponding to the cut-off at 75 per cent. of the stroke, that point relating to No. 6 of the series with varying travel and varying lead, and No. 4 of that with varying lap, in all of which examples the valve has the same proportions. The curves on this figure clearly show that with the shorter cut-offs varying lap affords the longest periods of expansion, but that it is in this respect nearly equalled by varying travel. They also show that when the cut-off is effected near the end of the stroke, the greatest amount of expansion is permitted by varying lead; and this is a fact to which we shall further allude presently.

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* This diagram is similar to those introduced for the same purpose in Mr. D. K. Clark's work *On Railway Machinery*.

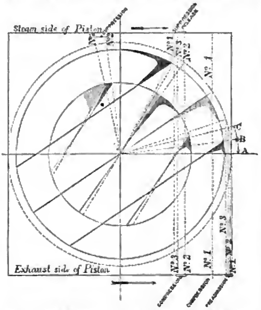
As regards the pre-admission, or admission of steam against the piston before the end of its stroke, there is little to choose between the second and third methods of altering the cut-off: for although with the higher grades of expansion this pre-admission is in the examples before us slightly greater with varying travel than with varying lap, yet it may, with the proportions of valves in ordinary use, be considered equal; and it can in both cases be easily increased or diminished by altering the lead.

When increase of lead alone, however, is employed to vary the expansion, the case is different—the pre-admission increasing so rapidly as to render, as before remarked, the amount of expansion practically attainable by this means very limited. It will nevertheless be seen from Fig. 130 that, with the proportion of valve there represented, the amount of expansion may be altered to the extent of about 10 per cent. by increasing the lead, without the pre-admission becoming at all excessive. Although in the preceding remarks we have to a certain extent discussed the positions in which the various proportions of valves enumerated in the Table stand with relation to the fourth and fifth qualifications of expansion gear, given on page 148, yet it must be remembered that, as before stated, the only real evidence on these points is that given by the indicator; and we shall therefore refer to that portion of this work, in which the use of that instrument is considered, for further information on these points. The sixth qualification also relates more to the action of the connecting rod, and of the gear by which the valve is driven, than to the proportions of the valve itself; and we may therefore leave it for future consideration, and proceed to investigate the action of lead and travel when varied simultaneously, the lap remaining constant.

4th. Lap constant, lead and travel varying.—When we arrive at that part of our subject which relates to the link motion, we shall find that, with some varieties of that valve gear, diminution of the travel of the valve is accompanied by an increase of the lead; and it is now our intention to show how far the distribution of steam thus effected differs from that produced by diminished travel alone. In Fig. 135 is shown,—*first*, the points of cut-off, exhaust, &c., corresponding to a valve having 5¼ inches travel, 1¼ inches lap, and ¼ of an inch lead; *secondly*, the alteration in the position of these points which is produced by a reduction of the travel to 3¾ inches, the lap and lead remaining the same; and, *thirdly*, the slight further modification which occurs when this decrease of the travel is accompanied by an increase of the lead from ¼ to ⅜ of an inch. The positions of the points of cut-off and release corresponding to these three cases, measured from the diagram, are as follows:—

		Per cent. of the Stroke.		Per cent. of the Stroke.
No. 1	Steam cut off at	75	Steam released at	91·9
2.	" "	45	" "	79·25
3.	" "	40	" "	75·25

It will be seen from this Table that the distribution of the steam effected by No. 3 varies very slightly from that which would be produced by a further slight reduction of the travel of No. 2; in fact, it may practically be considered as identical. The pre-admission is certainly greater in the case of No. 3; but this can scarcely be considered a disadvantage, as the use of the higher grades of expansion is, in locomotives, almost always accompanied by high speed of piston, and the increased pre-admission therefore becomes useful, as it permits the free access of the steam to the cylinder at the commencement of the stroke. Altogether, therefore, it may be concluded that the effect of decreased travel accompanied by increased lead is, if this increase is not excessive, practically the same as that of decreased travel alone. There is another fact shown by Fig. 135, to which it is necessary that we should allude; and that is the great increase in the *duration* of the pre-admission (measured in fractions of the revolution of the crank shaft) produced by a reduction in the travel of the valve, even when the lead is unchanged. An inspection of the diagram will show that the arc passed through by the crank pin after the opening of the port for the pre-admission of steam is when the travel of the valve is 3¾ inches, nearly twice as long as that described when the travel is 5¼ inches, the arcs being A C and A B respectively. If, therefore, the crank shaft revolved with the same velocity in both cases, the *time* allowed for the entrance of the steam into the cylinder before the commencement of the return stroke would with No. 2 be nearly twice as long as that given by No. 1.



No. 1.—Travel, 5¼ inches; . . Lap, 1¼ inches; . . Lead, ¼ inch.
" 2. " 3¾ " . . " 1¼ " . . " ¼ "
" 3. " 3¾ " . . " 1¼ " . . " ⅜ "

Fig. 135.

5th. Lap and travel varying, lead constant.—Those parts of the present chapter treating of varying lap and varying travel will have shown that, within certain limits, two valves having different amounts of lap and lead may, by proportioning their travels, be made to cut off the steam at the same point of the stroke. This being the case, we shall now proceed to investigate more minutely the actions of two such valves, in order to determine the theoretical advantages of a valve having great lap and long travel, as compared with one having a smaller amount of lap and proportionately less travel. The examples chosen for comparison are,—No. 1, lap, 1¼ inches; lead, ¼ of an inch; extreme travel, 5½ inches: and No. 2, lap, ⅝ of an inch; lead, ¼ of an inch; and extreme travel, 3 inches.

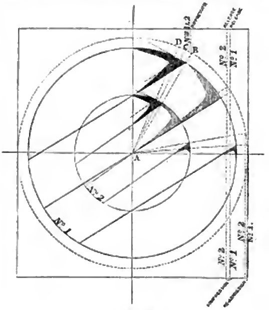


Fig. 136.

The action of these two valves when worked at their full stroke is shown on Fig. 136; from which diagram it will be seen that the steam is cut off at the same point by both,—viz., at 75 per cent. of the stroke. The release, and consequently the compression, occurs rather earlier with No. 2 than with No. 1, and the pre-admission also takes place sooner with the former valve. In Fig. 137 the valves are shown with their travels both reduced, so that they cut off the steam at 52½ per cent. of the stroke, that of No. 1 being made 4 inches, and that of No. 2, 2$\frac{3}{16}$ inches. In this, as in the previous instance, the valve No. 1 has the advantage of a slightly later release and compression; and it will be seen by a comparison of Figs. 136 and 137 that this advantage is increased with the earlier cut-off. Fig. 138 defines the limits of expansion attainable by each of the valves, the travels being in this instance reduced to twice the lap and lead, or 3 inches and 1¾ inches respectively, the angular advance of the eccentrics becoming 90°. It has been before stated that the shortest period of admission obtainable under these circumstances, from any valve, forms the same fraction of the whole stroke of the piston that the lead of the valve does of its travel; and, applying this rule in the present instance, we find that for No. 1 the earliest cut-off will take place at $\frac{\cdot 25 \times 100}{3} = 8{\cdot}33$ per cent. of the stroke of the piston; while for No. 2 it will occur at $\frac{\cdot 25 \times 100}{1{\cdot}75}$ $= 14{\cdot}28$ per cent.

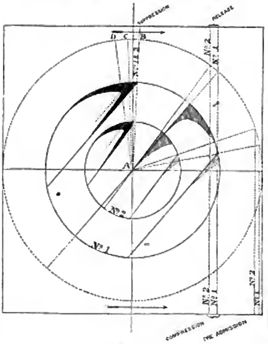


Fig. 137.

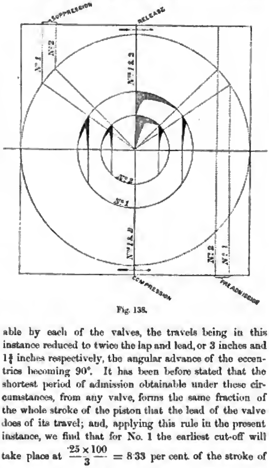


Fig. 138.

There is also another point by which the action of the two valves may be compared, and that is the relative *suddenness* of the cut-off effected by them. The performances of the two valves in this respect may be best compared by ascertaining the distance passed through by the crank, whilst the valves are changing from a position in which the steam ports are open to a small definite distance to that in which they are just closed. A convenient opening for the purpose of comparison is ⅛ of an inch; and, referring to Fig. 136, we find that with No. 1 the crank changes from the position A C to that marked

A B, whilst the valve is moved this distance; whereas, in the case of No. 2, the crank pin passes through the much greater arc D C B before the port is closed. With the reduced travel shown on Fig. 137 the result is similar, the suddenness of the cut-off estimated in this manner being in both instances in inverse ratio to the travel of the valve, as the angular movement of the crank is of course equal to the angular movement of the eccentric necessary to produce the traverse of the valve through the distance fixed upon. The valve with the longest lap and travel also gives a wider opening to the steam port, when working with any given degree of expansion, than that with the lesser proportions.

The foregoing observations tend to prove that the distribution of the steam effected by a valve having large lap and long travel is theoretically superior to that produced by a valve having a smaller amount of lap and proportionately less travel. Practically, however, the latter has the advantage of requiring less power to work it, accompanied by a corresponding diminution of wear and tear. The diminished opening of port and suddenness of cut-off due to the valve of the smaller proportions may be partially remedied by an increase in the length of the ports, but this valve, however, will still have the disadvantage of its action being more disarranged by a certain amount of wear in the joints of the valve gear, than that of one having greater travel would be under similar circumstances. These, however, are matters which will be more fully discussed presently.

Inside Lap and Clearance.—In almost all the valves hitherto considered in this chapter the proportions have been such that there has been no inside lap or clearance, and the action of these two elements therefore still remains to be considered. Inside lap shortens the period of exhaust, both by deferring the release and hastening the compression, while with inside clearance there is of course a small portion of the stroke of the piston during the performance of which both ports are uncovered to the exhaust. These actions were explained when describing Fig. 128, when it was shown that the effect of giving the valve there represented an amount of inside lap equal to *d f* would be to delay the exhaust to the extent N *h*, and hasten the compression by the equal quantity R *i*—the point N or R being that at which release and compression would occur if the valve was without inside lap or clearance. If, on the contrary, the valve had inside clearance equal to *e d*, the release and compression would commence at the points *g* and *k* respectively; and whilst the piston was travelling from *g* to *k*, or, what is the same thing, from *g* to *h*, both ends of the cylinder would be in communication with the exhaust. It will be evident, from what has been already stated concerning the action of outside lap, that the influence of any given amount of inside lap or inside clearance will be considerably increased as the travel of the valve is reduced; but the amount of either given to the valves of locomotive engines is usually so small that it is very doubtful if it practically affects the economical working of the steam, the gain at one part of the stroke being approximately balanced by the loss at another.

Before leaving this portion of the subject, it is advisable to show how the proportions of a valve necessary to produce any given amount of expansion can be determined geometrically. In practice this problem will generally present itself in one of the following forms:—1st. Lead and travel being fixed upon to determine the amount of outside lap and inside lap or clearance necessary to effect the cut-off and release of the steam at certain given points in the stroke of the piston; and, 2d. The lead and lap being given to find the extent of travel requisite to effect the suppression of the steam at a point fixed upon. These two cases shall be considered separately.

1st. *Lead and travel being fixed upon to determine the amount of outside lap and inside lap or clearance necessary to effect the cut-off and release of the steam at certain given points in the stroke of the piston.*—In Fig. 139 let the line A B (and therefore *a b* and *c d*, which are equal, parallel, and opposite to A B) represent, to different scales, the stroke of the piston and the travel of the valve, and let also C and D be the points in the former at which it is desired that the suppression and release of the steam should take place. Bisect A B, and from the point of bisection, E, describe a circle with a radius equal to E A. From the points C and D let fall the lines C *g* and D *h* perpendicular to A B; these will intersect the circle just drawn at the points F and H. From the point A as a centre, and with the given amount of lead as a radius, describe an arc, *e f*, and draw a line, G F, touching this arc and passing through the point F already obtained. The distance, E *i*, of this line (measured at right angles to it) from the centre, E, of the circle is equal to the amount of lap necessary to effect the desired cut-off of the steam. Next draw through the point H another line, H I, parallel to F G; if, as in the figure, this line passes through the centre, E, of the circle, no inside lap or clearance is required to cause the steam to be released at the point D in the stroke of the piston. If, however, it does not so pass through the centre, its distance from it, measured in the same way as that of the line F G for outside lap, will indicate the amount of inside lap or clear-

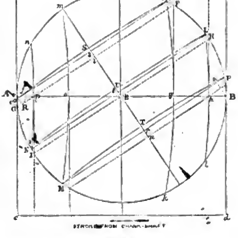


Fig. 139.

ance necessary, the latter being required if this line, H I, falls on the same side of the centre as F G does.

The lap thus determined would, if applied to both ends of the valve, of course effect the cut-off of the steam at the points desired during both strokes of the piston, if it were not for the disturbing influence of the connecting rod, which causes the distance traversed by the piston before the suppression occurs to be longer when the piston is moving towards the crank shaft than when its motion is in the opposite direction. To correct the irregularity thus produced, the lap may, as has been previously stated, be increased at one, and decreased at the other edge of the valve; and the amount of increase and decrease necessary to equalize the admissions of the steam during both strokes may be easily determined by a slight modification of the diagram, as follows:—With the length of the connecting rod (taken to same scale as that to which A B represents the stroke of the piston) as a radius, and from a centre situated on the line A B produced, draw arcs, K *g k* and L *h l*, passing through the points *g* and *h*, and also similar ones, M *o m* and N *p n*, passing through the points *o* and *p*, which mark the positions of the piston at which the suppression and release of the steam are to occur during its stroke from B to A. All these arcs are to have their *convex* sides next the crank shaft end of the stroke. Next, through the points K, L, N, and M, at which these arcs cut the circle, draw lines, K O, L N, and M P, parallel to H I, then the distance of these lines from the centre, E, will give the corrected laps for both ends of the valve.

In Fig. 139, E *r* represents the amount of lap necessary upon the edge of the valve next the crank shaft, and E *s* that required at the other edge. On the inside the valve will require, at the crank-shaft end, a lap, and at the other end a clearance equal to E *t*. In practice, the irregularity in the release of the steam, caused by the connecting rod, is very frequently neglected, and the cut-off only is equalized by the use of unequal lap. An inspection of the diagram will show, however, that a better distribution of the steam can be obtained by increasing the length of the eccentric rod by the amount E *t*, and keeping the laps of the valve equal; the effect of this being that, when the eccentric is at half-stroke, the valve, instead of being at the centre of its travel, will be removed from it by the distance E *t*, and the lap at the crank-shaft end will be decreased, and that at the other end increased by that amount. The difference in the laps thus produced will vary from the correct one determined above only by the amount of the difference between E *t* and *s i*, or *r u*, a difference which is practically of no importance when the cut-off does not occur very early in the stroke; the variation in the inside lap and clearance will, of course, be exactly correct. If, on the other hand, the eccentric rod be lengthened by an amount which is a mean between the distances *s i* and *r u*, an almost exactly equal cut-off will be obtained during both strokes of the piston, at the expense of a very slight irregularity in the release; and altogether this mode of equalizing the distribution of the steam appears to be far preferable to the employment of unequal lap.

Whether unequal lap or increased length of eccentric rod be employed for correcting the irregularities produced by the action of the connecting rod, unequal lead is a necessary result, the extent of the inequality being in the one case equal to the difference of laps, and in the other to twice the amount by which the distance from the centre of the eccentric to the centre of the valve exceeds that from the centre of the crank shaft to the centre of the exhaust port. In the diagram (Fig. 139) it will be seen that, whereas when the lap was the same at both edges of the valve, the lead was equal to A R; the employment of unequal lap increased it to B *w* at the crank-shaft end of the stroke, and decreased it to A *v* at the other.

For all practical purposes the differences of lap, &c., determined as above described, are sufficiently correct; but if minute accuracy is required, they must be subjected to a further correction due to the action of the eccentric rod. The amount of alteration necessary to compensate for this action can be determined by substituting for the straight lines K O, M P, and N L, arcs passing through the points at which these lines cut the circle, and drawn with a radius equal to the length of the eccentric rod, taken to the same scale as that to which the line A B represents the travel of the valve. These arcs are drawn so as to increase the larger and diminish the smaller lap, which will thus become equal to E S and E T respectively, the inside lap and clearance being increased to E U.

2d. *Lap and lead being fixed to determine the amount of travel necessary to effect a given admission of the steam.*—For the solution of this problem we must proceed as shown by Fig. 140. Draw the line A B, representing to any convenient scale the stroke of the piston —bisect it,—and from the point of bisection C, as a centre, with a radius equal to C A, describe a circle A E B. Upon the line A B mark the point of the stroke at which it is desired that the steam should be cut off—for instance, at D; and at this point erect a perpendicular to the line

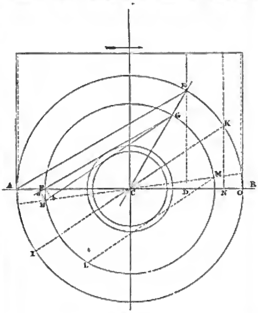


Fig. 140.

A B, which will cut the circle at a point, E. From the centre, C, describe also two other circles, the radius of one being equal to the given lap, and that of the other equal to the lap added to half the lead. The scale employed for these need not be the same as that used for the first circle, but may, as in the figure, be larger with advantage. Join C E and A E, and draw the line F G parallel to that uniting the two latter points, and touching the larger of the two circles last described. From the same centre, C, describe a fourth circle, passing through the point, G, at which the line, F G, just drawn cuts that marked C E; this circle will also pass through the point, F, at which F G and A B intersect each other, and its diameter will equal the travel of valve required. The travel thus determined is not the mathematically correct one, but it is sufficiently exact for all practical purposes. To determine the amount of error, draw from the point G the line G H, touching the circle of which the radius is equal to the lap, and it will then be seen that the lead given by this travel, when the cut-off takes place at the desired point, is less than that required by the terms of the problem to the extent of the difference between the length of the lines *a* H and F *b*, an amount which is quite insignificant with all ordinary proportions of valves. By drawing the lines I K and L M parallel to G H, the diagram becomes similar to that described in the latter part of the previous chapter, and the points of the stroke, N and O, at which the exhaustion and pre-admission of the steam will respectively occur, with the proportion of valve and travel under consideration, can be determined as then explained.

If instead of the lead of the valve being given, the point of the stroke of the piston at which the pre-admission shall commence be fixed upon, the diagram for determining the necessary travel of the valve will assume the form shown by Fig. 141. In this as in the previous instance, the line A B represents the stroke of the piston, and the circle A E B F is described from the centre, C, of the line A B, with a radius equal to half its length. The point D is that at which the steam is to be cut off during the stroke of the piston from A to B, and G is that at which the pre-admission is to commence during the stroke in the opposite direction. From the points D and G draw perpendiculars to the line A B, which will cut the circle at the points E and F. Join E C, C F, and E F. From the centre, C, describe a circle with the given lap as a radius, and draw the line H I parallel to E F, and touching this circle. The distance (H C or I C) from the centre, C, of the points at which this line intersects those marked E C and F C is equal to the radius of a circle of which the diameter is the travel required. Draw this circle, and the distance K L will then be equal to the amount of lead with which the valve must be set to give the pre-admission fixed upon. The travel determined by this diagram is the exact one, and not an approximation, as in the previous instance. In both cases the scale employed for measuring the lead and travel must be the same as that to which the lap is laid down; but, as has been already mentioned, it need not be identical with that employed for representing the stroke of the piston.

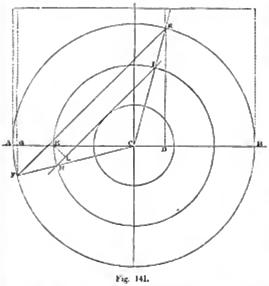


Fig. 141.

Instead of employing geometrical means for ascertaining the distribution of the steam effected by a slide valve driven by a single eccentric, we may readily determine it by a few simple trigonometrical and arithmetical calculations; and for this purpose the following rules will be found useful.

All the trigonometrical formulæ are founded upon the simple fact that, as the crank and eccentric are both fixed upon the crank shaft, the angle contained between lines drawn through their centres from the centre of the crank shaft is a constant one, and thus, if the position of the eccentric is known, the position of the crank can be determined, and *vice versa*. It has also been shown in the course of this and the preceding chapter that, if the positions of the centres of the crank pin and eccentric are known, the positions of the piston and valve corresponding to them can be obtained (if the disturbances caused by the varying angularity of the connecting and eccentric rods be disregarded) by means of ordinates drawn from the respective centres to the centre line of the motion; and, combining these facts, it follows that, if the position of the valve be given, the position of the piston corresponding to it can be calculated, or, inversely, that the position of the valve agreeing to any given position of the piston can be obtained.

The first thing to be done is to ascertain the angle contained between the centre lines of the crank and eccentric. To do this it is only necessary to divide the linear advance of the eccentric by the half travel, when the quotient will be equal to the natural sine of the angular advance (B, Fig. 123, page 139), and opposite this number in a table of natural sines will be found the corresponding angle. The total angle between the centre lines of the crank and eccentric will be evidently equal to the angle so obtained, added to 90°. In the subjoined Table are given the angular advances (to the nearest twelfth of a degree) due to various proportions of linear advance and travel, and by reference to it the angular advance corresponding to most of the proportions of linear advance and travel in ordinary use can be determined at sight. It will also be found useful for carrying out some other calculations, as will be explained presently.

TABLE SHOWING THE ANGULAR ADVANCE CORRESPONDING TO VARIOUS PROPORTIONS OF LINEAR ADVANCE AND TRAVEL.

Travel in Inches.	Linear Advance in Inches.															
	⅜	½	⅝	¾	⅞	1	1⅛	1¼	1⅜	1½	1⅝	1¾	1⅞	2	2⅛	2¼
6½	6°40′	8°50′	11° 5′	13°20′	15°35′	17°55′	20°15′	22°35′	25° 0′	27°30′	30° 0′	32°35′	35°15′	38° 0′	40°50′	43°50′
6⅜	6 45	9 0	11 20	13 35	15 55	18 15	20 40	23 5	25 35	28 5	30 40	33 20	36 0	38 50	41 50	45 25
6¼	6 55	9 15	11 35	13 55	16 15	18 40	21 5	23 35	26 5	28 40	31 20	34 5	36 55	39 50	42 50	46 5
6⅛	7 0	9 25	11 45	14 10	16 35	19 5	21 35	24 5	26 40	29 20	32 0	34 50	37 45	40 50	43 55	
6	7 10	9 35	12 0	14 30	17 0	19 30	22 0	24 35	27 15	30 0	32 50	35 40	38 40	41 50	45 5	
5⅞	7 20	9 50	12 15	14 45	17 20	19 55	22 30	25 10	27 55	30 40	33 35	36 35	39 40	42 55		
5¾	7 30	10 0	12 35	15 5	17 45	20 20	23 0	25 45	28 35	31 25	34 25	37 30	40 40	44 5		
5⅝	7 40	10 15	12 50	15 30	18 10	20 50	23 35	26 25	29 15	32 15	35 20	38 30	41 50	45 20		
5½	7 50	10 30	13 10	15 50	18 35	21 20	24 10	27 5	30 0	33 5	36 15	39 30	43 0			
5⅜	8 0	10 45	13 25	16 10	19 0	21 55	24 45	27 45	30 45	33 55	37 15	40 35	44 15			
5¼	8 15	11 0	13 50	16 35	19 30	22 25	25 25	28 25	31 35	34 50	38 15	41 50	45 35			
5⅛	8 25	11 15	14 5	17 0	20 0	23 0	26 5	29 10	32 25	35 50	39 20	43 5				
5	8 40	11 30	14 30	17 30	20 30	23 35	26 45	30 0	33 25	36 55	40 35	44 30				
4⅞	8 50	11 50	14 50	17 55	21 0	24 15	27 30	30 50	34 20	38 0	41 50	45 55				
4¾	9 5	12 10	15 15	18 25	21 35	24 55	28 15	31 45	35 20	39 10	43 10					
4⅝	9 20	12 30	15 40	18 55	22 15	25 40	29 5	32 45	36 30	40 25	44 40					
4½	9 35	12 50	16 5	19 30	22 50	26 25	30 0	33 45	37 40	41 50	46 15					
4⅜	9 50	13 15	16 35	20 5	23 35	27 10	30 55	34 50	38 55	43 20						
4¼	10 10	13 35	17 5	20 40	24 20	28 5	32 0	36 0	40 20	44 55						
4⅛	10 30	14 0	17 40	21 20	25 5	29 0	33 5	37 20	41 50	46 40						
4	10 50	14 30	18 15	22 0	25 55	30 0	34 15	38 40	43 25							
3⅞	11 10	14 55	18 50	22 45	26 50	31 5	35 30	40 10	45 15							
3¾	11 35	15 30	19 30	23 35	27 50	32 15	36 55	41 50	47 10							
3⅝	11 55	16 0	20 10	24 25	28 50	33 30	38 20	43 35								
3½	12 20	16 35	20 55	25 25	30 0	34 50	40 0	45 35								
3⅜	12 50	17 15	21 45	26 25	31 15	36 20	41 50	47 45								
3¼	13 20	17 55	22 35	27 30	32 35	38 0	43 50									
3⅛	13 55	18 40	23 35	28 40	34 5	39 50	46 5									
3	14 30	19 30	24 35	30 0	35 40	41 50	48 35									
2⅞	15 5	20 20	25 45	31 25	37 30	44 5										
2¾	15 50	21 20	27 0	33 5	39 30	46 40										
2⅝	16 35	22 25	28 25	34 50	41 50	49 40										
2½	17 30	23 35	30 0	36 55	44 30											
2⅜	18 25	24 55	31 45	39 10	47 30											
2¼	19 30	26 25	33 45	41 50	51 0											
2⅛	20 40	28 5	36 0	45 25												
2	22 0	30 0	38 40	48 35												
1⅞	23 35	32 15	41 50	53 10												
1¾	25 25	34 50	45 35													
1⅝	27 30	38 0	50 20													
1½	30 0	41 50	56 25													

When using this Table, it must be remembered that the angular advance is determined by the *proportion* which the linear advance bears to the half travel, and that the dimensions representing these elements may be either both multiplied or both divided by any number without affecting the result. This being the case, it follows that the preceding Table is available for determining the angular advance corresponding to many dimensions of linear advance and travel which do not appear in the index columns. Thus, to find the angular advance due to a travel of 3 1/16 inches and linear advance of 11/16 inches, we have only to double these dimensions, when, opposite the travel 6⅛ inches in the Table, and under the linear advance 1⅜ inches, will be found 26° 40′, which is the angular advance required.

The position of the crank corresponding to any given position of the piston can be obtained by dividing the distance of the piston from the *centre* of its stroke by the throw of the crank, when the quotient will be equal to the natural sine of the angle contained between the centre line of the crank and a line drawn from the centre of the crank shaft perpendicular to the direction of the stroke of the piston, or, what is the same thing, to the cosine of the angle contained between the centre line of the crank and the centre line of the engine. We shall now apply these facts to the solution of the problems connected with the action of a slide valve and single eccentric.

The lap, lead, and travel of a valve being given, to determine the point in the stroke of the piston at which the steam will be cut off.—The suppression of the steam occurs as the valve is approaching the centre of its travel, but is removed from that centre by the amount of the lap; and the position of the eccentric corresponding to the position of the valve can thus be determined by dividing the lap of the valve by its half travel, when the quotient will equal the cosine of the angle contained between the centre line of the engine and a line drawn from the centre of the crank shaft through the centre of the eccentric.

Thus, in Fig. 142, let A be the centre of the crank shaft, and C B the direction of the stroke of the piston; and let also the larger dotted circle represent the path of the centre of the crank pin, and the smaller that of the centre of the eccentric. Also, let A D be the centre line of the eccentric when the valve has just closed the steam port (the distance *a* being equal to the lap), and E A the corresponding position of the crank. Then the lap divided by the half travel will equal the cosine of the angle B A D. Now, it has been shown above how the angle E A D, contained between the centre lines of the crank and the eccentric, can be determined, and if from this angle we subtract the angle B A D, we shall have as a remainder the angle E A B contained between the centre line of the crank and the centre line of the engine, at the instant that the steam is cut off.

and by multiplying the natural cosine of the angle by the length of the crank, the distance of the piston from the *centre* of its stroke can be obtained. If the angle E A B

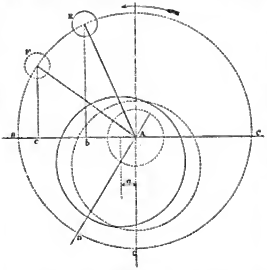


Fig. 142.

is less than a right angle, the distance that has been traversed by the piston, measured from the commencement of its stroke, will be equal to the half-stroke added to the distance just obtained; if, on the other hand, the angle E A B is greater than a right angle, the distance of the piston from the commencement of its stroke will be less than the half-stroke by a similar amount. Putting these results into the form of equations, and calling the angle B A D the angle of eccentric, we have—

$$\text{Cosine angle of eccentric} = \frac{\text{lap}}{\frac{1}{2}\text{ travel}}$$

Angle of crank = angle contained between crank and eccentric — angle of eccentric. Distance of piston from *centre* of stroke = length of crank × cosine angle of crank.

Instead of employing the above trigonometrical formulæ to ascertain the point at which the steam is cut off, we may use the following simple arithmetical rule,* which will be found preferable in practice:—Add the lap of the valve to half the lead, and divide the sum by the half-travel. Square the quotient, and multiply it by the stroke of the piston,—the product will be equal to the distance of the piston from the *end* of the stroke when the suppression occurs. All the dimensions should be expressed in the same unit of measurement. Algebraically stated, the rule will stand thus:—

Distance of piston from end of stroke when steam is

$$\text{cut off} = \left(\frac{\text{lap} + \frac{1}{2}\text{ lead}}{\frac{1}{2}\text{ travel}}\right)^2 \times \text{stroke of piston.}$$

This rule, however, is not applicable unless the suppression takes place late in the stroke.

To determine the point of release,—the lap, inside lap or clearance, lead, and travel of the valve being given.—If a valve has neither inside lap nor clearance, the release of the steam takes place when the valve is at half-stroke, and the angle which the crank forms with the centre line of the engine is then equal to the angular advance of the eccentric; the distance of the piston from the centre of its stroke being equal to the cosine of the angular advance of the eccentric, multiplied by the length of the crank. In Fig. 142, for instance, the dotted lines show the positions of the crank and eccentric at the point of release,—the angle of crank, F A B, being equal to the angular advance, and the distance A c being equal to the cosine of the angular advance multiplied by the length of the crank.

If the valve has inside lap or inside clearance, the calculation becomes very similar to that for determining the point of suppression, the eccentric being removed from the centre of its travel by the amount of such inside lap or clearance, when the release of the steam takes place. The sine of the angle which the centre line of the eccentric then forms with the line A G (Fig. 142) is equal to the inside lap or clearance divided by the half-travel, and from this the angle can be ascertained. If the valve has inside lap, the angle thus found must be subtracted from,—or, if it has inside clearance, it must be added to—the angle of crank, F A B, determined by the preceding rule; and the cosine of the remainder or sum, as the case may be, must be multiplied by the length of the crank in order to obtain the distance of the piston from the centre of its stroke. Thus with inside lap the formula becomes—

Distance of piston from centre of stroke = cosine (angular advance — angle due to inside lap) × length of crank.

Whilst for inside clearance it is—

Distance of piston from centre of stroke = cosine (angular advance + angle due to inside clearance) × length of crank.

In all cases, unless the sum of the linear advance of the eccentric and inside clearance exceeds the half-travel of the valve (and such a case seldom occurs in practice), the distance of the piston from the centre of its stroke, determined by the above formulæ, is to be added to the half-stroke to get the distance traversed by the piston from the commencement of its stroke at the time the release takes place.

The point of release, like that of suppression, can also be determined by arithmetical means, the rule, if the valve has neither inside lap nor clearance being as follows:—Add the lap to the lead and divide the sum by the half-travel. Let the quotient be called x, and the distance of the piston from the end of its stroke when the exhaust opens will then be equal to $\left\{1 + \sqrt{(1 + x)(1 - x)}\right\} \times$ half-stroke of piston. The distance of the piston from the end of the stroke will be given in terms of the same unit of measurement as that used to express the "half-stroke of piston" in the equation; if, for instance, the position of the piston be required in per centages of the stroke, the equation becomes

Distance of piston from end of stroke when exhaust opens

$$= 50\left\{1 + \sqrt{(1 + x)(1 - x)}.\right\}$$

When the valve has inside lap or clearance the formula becomes more complex, and the problem can then be more readily solved by the trigonometrical method.

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* This rule, as well as that given a little farther on, for determining the point of release, was, we believe, first published by Mr. J. M'Farlane Gray, whose Great-Circle Diagram we described in the last chapter.

To determine the point of pre-admission.—The pre-admission of the steam will take place when the valve is moving *from* the centre of its stroke, and is removed from that centre by the amount of the lap. The angle contained between the centre line of the eccentric and the vertical line A G (Fig. 142) will then be equal to the angle D A G, but will be on the other side of the line A G. The sine of this angle is equal to the lap of the valve divided by the half-travel. Call this the angle due to lap; then, angle of crank = angular advance — angle due to lap; and the distance of the piston from the end of its stroke = length of crank — $\left\{\text{cosine (angular advance of eccentric} - \text{angle due to lap)} \times \text{length of crank.}\right\}$

To find the point at which compression commences.—If the valve has neither inside lap nor clearance, the compression will commence at the one side of the piston at the same time as the exhaust opens on the other, and the same rule as was used for determining the point of exhaust will of course apply. If the valve has inside clearance, the compression will begin at the same point as the exhaust would take place if the valve had an equal amount of inside lap, and *vice versa*. To obtain the position of the point of compression, it is therefore only necessary to reverse the formulæ already given for determining the point of exhaust. Thus, with inside clearance, the distance of the piston from the centre of the stroke when compression commences = cosine (angular advance of eccentric — angle due to clearance) × length of crank. If the valve has inside lap, the rule becomes—Distance of piston from centre of stroke = cosine (angular advance of eccentric + angle due to inside lap) × length of crank.

After valve gear has been some time in use, the wear of the joints alters to some exent the distribution of the steam effected by it. The effect of a slight amount of wear is, however, not so great as is commonly supposed. In Fig. 143 we show the proper distribution effected by a valve having 1¼ inches lap, ¼ inch lead, and 5½ inches travel, and also the alteration in this distribution caused by a looseness of ¼ inch in the joints of the valve gear. This looseness causes the valve to be stationary for a short period at each end of the travel, as shown by the dotted lines. By this action the suppression and exhaustion of the steam are both deferred; in our example they take place respectively 1½ per cent. and 1 per cent. of the stroke later than when the gear is working without slack, whilst the lead is of course reduced by half the amount of the wear. The effect of a certain amount of wear is of course more marked in the case of short than of long travels; but in any case, although it reduces the stroke of the valve, its action on the distribution of the steam is very similar to that which would be produced by an increase of travel.

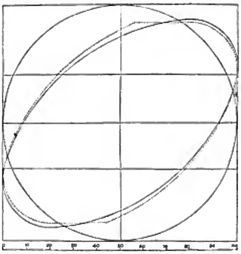

Fig. 143.

CHAPTER VIII.

LINK MOTION—THE SHIFTING LINK.*

The locomotives of the present day, instead of having the slide valve driven directly from the eccentric, as was the case in the older engines fitted with the "gab motion," have interposed between the eccentrics and valve, certain descriptions of gear, by the employment of which the motion imparted to the slide valve can be modified in various ways. The valve gears thus used may be divided into two classes, the first comprising those gears in which one or more eccentrics *fixed* on the driving axle are employed, the variation in the motion of the valve being regulated by link work introduced between it and the eccentrics; and the second consisting of those in which the position of the eccentric or eccentrics on the driving axle is capable of alteration. In the first of these divisions are included the "link motions" commonly so called, and it is to these that we intend first to direct our attention.

In the link motions in general use two eccentrics are employed, the position of these on the driving axle being such that one can give to the valve the motion required when the engine is moving forward, whilst the other furnishes the means of moving in the opposite direction. These link motions may themselves be classified as follows:—*First*, "Shifting link" motions, or those in which the curved link connecting the ends of the eccentric rods is movable vertically, being supported by links from the end of the reversing arm, while the sliding block by which the motion is transferred from the link to the valve spindle is subjected to horizontal motion only. *Second*, "Stationary link" motions, or those in which the expansion link is supported from a fixed point in the engine frame, the block sliding in the link being carried by a radius rod attached to the valve spindle, and having its position adjustable vertically by links from the end of the reversing arm; and *Third*, Those in which the two preceding varieties are combined, both the expansion link and the block being capable of being moved vertically. Examples of these three classes are furnished by the London and North-Western engine "Lady of the Lake" (Plate II.), the Great Eastern express engine (Plate IX.), and Messrs. Beyer & Peacock's express engine (Plate III.), respectively.

These link motions have to act in the twofold capacity of reversing and expansion gears, the first of these objects being of easy attainment, whereas the adaptation of the link motion to the latter purpose demands considerable care and attention in the proportioning and arrangement of its parts. The properties which a good expansion gear ought to possess having been enumerated in the last chapter, it will be unnecessary to repeat them here; and we shall therefore at once proceed to investigate the action of link motions belonging to the first of the above-mentioned classes.

The Shifting Link.—This valve motion consists of the following parts (see Fig. 144):—

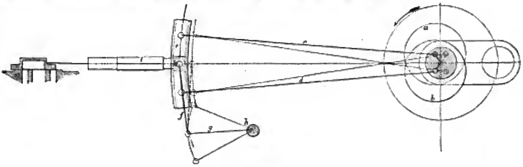

Fig. 144.

1. The two eccentrics; *a*, the fore-gear, and *b*, the back-gear eccentric.

2. The fore-gear and back-gear eccentric rods, *c* and *d*, by which the motion of the eccentrics is transmitted to the link.

3. The expansion link, *e*, generally known by the name "link" simply.

4. The lifting link, *f*, connecting the expansion link with the end of the lifting arm by which its position is regulated.

5. The reversing or lifting arm, *g*, keyed on to the weigh-bar or reversing shaft, *h*.

6. The valve spindle, *i*, carrying the block, *k*, sliding in the link. The valve spindle is in the diagram represented as movable in the direction of the centre line of the motion only, it being maintained in that line by fixed guides. This, however, is not always the case, part of the spindle being sometimes jointed and carried by links attached to some fixed point in the framing of the engine. The effect of these varieties of construction will be illustrated afterwards.

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* By Mr. W. H. Maw, *vide* footnote, p. 139.

The motion is said to be in fore or back gear according to whether the joint of the fore or back-gear eccentric rod is nearest the centre of the block.

The first point in this motion which claims our attention is the form of the expansion link by which the ends of the two eccentric rods are connected. The principal varieties in ordinary use are represented by Figs. 145, 146, 147, and 148. These differ, as will be seen, not only in their mechanical construction, but also in the relative positions occupied by the points at which the eccentric rods and lifting links are attached. The "box link" shown by Fig. 145 is formed in halves bolted together at the top and bottom, the inner sides of the halves being recessed for the reception of the sliding blocks carried by the end of the valve spindle. On the outside of the halves are formed studs for the attachment of the eccentric rods and lifting link. When this form of link is employed, these studs are generally arranged as shown on the figure—viz., on the centre line of the link—that for the lifting link being equidistant from the other two. This is not, however, always the case.

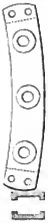
Fig. 145.

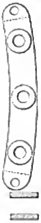
Fig. 146.

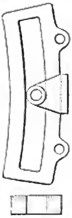
Fig. 147.

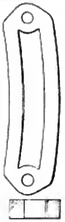
Fig. 148.

Fig. 146 represents a modification of the box link just described, the sliding blocks being in this case recessed instead of the sides of the link. In both these links, it will be observed that the blocks can be traversed throughout their entire length between the end distance pieces, and the centre of the block can thus be made to coincide with the centre of the joint of either of the eccentric rods, or even be moved beyond them if required.

The "open link" is made in one piece, the block sliding in a slot cut in it. In the form shown by Fig. 148 (which is that originally employed by Stephenson) the eccentric rods are coupled to the ends on the centre line of the link, and the joint of the lifting links coincides with that of one of the eccentric rods. In this link the block is of course always between the ends of the eccentric rods, and therefore, as will be presently seen, the full travel of the eccentrics cannot be communicated to the valve. In the link shown by Fig. 147 this defect is obviated;—the eccentric rods being coupled behind, allow the block to be traversed past them if necessary. The centre of suspension, or that to which the lifting links are attached, is, in the example before us, shown placed upon the centre line of the link at the centre of its length. It is, however, sometimes placed behind, like the knuckle of the eccentric rods, sometimes at the end, as in the link last described, and sometimes in a position intermediate between these. Each of the varieties of link just mentioned has its own peculiar effect upon the motion of the valve, and the action of each must therefore be investigated separately. Before doing this, however, it is necessary that we should ascertain the correct position of the two eccentrics, and then determine the nature of the motion transmitted to the link by their combined action. In the following preliminary investigation we shall suppose the form of link known as the "box link" to be employed, the centre of suspension and points of attachment of the eccentric rods being arranged as shown on Fig. 145.

The Position of the Eccentrics.—When the valve motion is in full gear, the centre of the studs of one of the eccentric rods coincides with that of the sliding block, so that the position which the eccentric that is in gear should occupy with respect to the crank should be the same as if that eccentric alone was employed to work the valve—the linear advance being equal to the lap *plus* the lead. This reasoning, of course, applies to both eccentrics. When, however, the link is placed in mid gear, or in any intermediate position between that and full gear, we shall find that the valve will possess a greater or less amount of lead than it had when in full gear, according to the manner in which the eccentric rods are attached to the link. Generally the rods are disposed as shown in Fig. 144, so that they cross when the centres of the eccentrics are on that side of the driving axle which is farthest from the link, and by this arrangement the lead of the valve is increased as the link approaches mid gear.

The way in which this increase of lead is produced will be best explained by tracing out the movements which occur when the link is changed from full to mid gear. For this purpose we refer to Fig. 151, in which A B represents the centre line of the valve motion, C the centre of the driving axle, D the centre of the fore-gear, and E that of the back-gear eccentric, the crank being in the position C B. When the link is in full fore gear, the centre of the joint of the fore-gear eccentric rod, where it is coupled to the link, is at F, and that of the back-gear rod at K, the valve being at the same time in the position shown by the dotted line, G, at the top of the diagram, and the steam port being uncovered by the amount of the lead, G H, which is equal to the linear advance (E I) of the eccentrics, *minus* the lap of the valve. If we now suppose the link to be raised into mid gear, we see that a compound movement takes place, the end of each eccentric rod describing an arc around the centre of its own eccentric, as shown by the lines F *f*, K *k*. The effect of this movement is that the centre line of the expansion link intersects the centre line of motion at a greater distance from the driving axle than when in full gear, and the centre of the block is consequently removed to the position *a*, and the lead is thus increased by the amount F *a*, the valve assuming the position *g*. The increase of lead thus produced might obviously be removed by lessening the curvature of the expansion link, were it not that at the opposite end of the stroke of the piston, the increase of lead caused by placing the link in mid gear would be augmented by such a procedure. This is proved by the dotted lines which represent the positions occupied by the various parts when the crank is situated at L C. Thus *d* is the centre of the fore-gear, and *e* that of the back-gear eccentric; M N the positions of the joints of the respective rods when the link is in full gear, and *m n* the positions when in mid gear. In this case it will be seen that the effect of placing the link in mid gear is to make the centre line of the link intersect the centre line of motion at a point nearer to the driving axle, and thus to cause an increase of lead, as at the other end of the stroke of the piston.

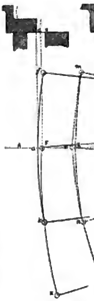

Fig. 149.

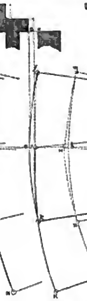

Fig. 150.

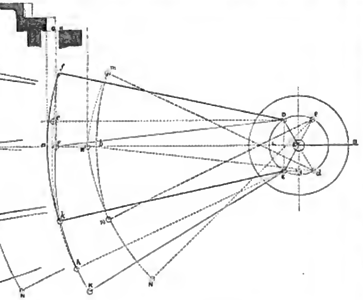

Fig. 151.

Under these circumstances it is evident that when the eccentric rods are disposed as shown in the diagram, it is impossible to avoid an increase of the lead of the valve, as the link is moved from full to mid gear; and all that can be done, therefore, is to render this increase equal for both ends of the stroke of the piston, or, in other words, make *a* F = M *b*. The equality or non-equality of these two distances manifestly depends upon the amount of curvature given to the expansion link; and they will be found to be practically equal when the centre line of the link is drawn with a radius equal to the length of the eccentric rods, measured from the centre of the eccentric to the centre of the joint at the other end. There are, however, some cases in which it is advisable to depart from this rule for the curvature of the link, and these will be explained hereafter when the various methods of adjusting this description of link motion are described.

It will be seen from Fig. 151 that when the link is being changed from fore to mid gear, the movements of both the eccentric rods tend to increase the lead until the link assumes the position *c h*, at which time the fore-gear rod *c* D becomes parallel to the centre line of the motion. After this point is passed, the lead is increased by the *difference* of the action of the two rods; and the variation, therefore, takes place more gradually as the link approaches mid gear. In the example given it will be seen that half the total increase has been caused by shifting the link from full gear to the position *c h*.

Although the increase of lead towards mid gear is unavoidable when this particular arrangement of the shifting link is employed, yet the amount of the increase is subject to variation by alteration in the proportions of the parts composing such valve gear. For instance, we have represented in Figs. 149, 150, and 151, the effects of alterations in the length of the eccentric rods which have, in these three diagrams, lengths equal to twenty-four, sixteen, and eight times the throw of the eccentrics respectively. As this latter dimension has been taken for example, as 2¼ inches, the lengths of the rods are 54, 36, and 18 inches, the distance between the points F and K on the expansion link being 12 inches. The letters of reference are the same in each case; and a comparison of these figures will show that the increase of lead in mid gear becomes less as the length of the eccentric rods is increased. A similar diminution is effected by shortening the distance between the points at which the eccentric rods are attached to the expansion link. In Fig. 152 the points A and B show the positions of the joints of the fore and back-gear eccentric rods attached to the expansion link A D B, the link being shown in mid gear. The points *a* and *b* similarly represent the positions the same joints would occupy if attached to the expansion link *a* E *b*, the distance between the centres *a* and *b* on this link (which is also placed in mid gear) being only equal to half that between the centres A and B on the link A D B. When the links are in full fore gear, the joint of the fore-gear eccentric rod would in either case occupy the position C, that of the back-gear rod being situated at B when attached to the short, and at *c* when coupled to the long expansion link. This diagram shows, therefore, that with the long expansion link the increase of lead towards mid gear would amount to D C, whereas with the shorter link it would be equal to E C only.

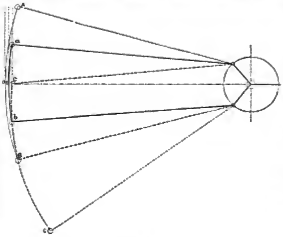

Fig. 152.

Another element which affects by its proportions the amount by which the lead is increased in mid gear, is the throw of the eccentrics. The influence thus exercised is shown by Fig. 153, in which let the points A and B represent the positions occupied, at the commencement of the stroke of the piston *from* the crank shaft, by the centres of a pair of eccentrics having 3 inches throw, and set with a linear advance of 1½ inches. Also, let the dotted line C D E show the position of an expansion link (12 inches between the end centres) in full gear, and connected with the eccentrics just mentioned, and the full line G F D represent the same link when in mid gear. Under these circumstances we see that the increase of lead in mid gear is equal to F C.

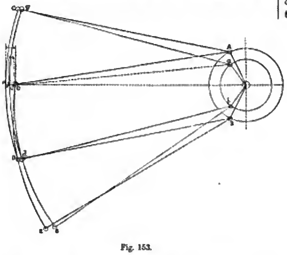

Fig. 153.

Let us now suppose the throw of the eccentrics to be reduced to 2 inches, the linear advance remaining the same, the position of their centres will then be represented by the points *a* and *b*. If we now couple to these eccentrics a link and rods of the same dimensions as those which were used with the eccentrics having 3 inches throw, we shall find that the link will assume, when in full and mid gear, the positions *c d e* and *g f d* respectively, the increase of lead in mid gear being only equal to *f c*—much less than in the former instance. It will be noticed that in the diagram the points C and *c* do not quite correspond;—this is owing to the difference in the obliquity of the eccentric rods, which, with the very short rods represented in the figure, exercises considerable influence. With the proportions of rod in ordinary use, however, the disturbance arising from this cause would be considerably less.

From the investigation of the foregoing diagrams we therefore find that, when the eccentric rods of a shifting link motion are arranged so as to cross as therein represented, an increase of lead takes place as the link approaches mid gear, but that the amount of such increase may be diminished,—*first*, by increasing the length of the eccentric rods; *second*, by diminishing the distance between the points at which the eccentric rods are attached to the expansion link; and *third*, by decreasing the throw of the eccentrics: in all these cases the dimensions of those parts of the motion which are not mentioned as being altered being supposed to remain the same.

We will now consider the change in the amount of the lead which takes place when the eccentric rods are coupled to the expansion link, so that they cross when the centres of the eccentrics are on that side of the crank axle which is nearest the link. In Fig. 154, in which the rods are so arranged, let A and B represent the centres of the forward and backward gear eccentrics when the crank occupies the position C D. Let the line E F G represent the position of the expansion link when in mid gear, and H G I that in full back gear, the fore-gear eccentric rod being coupled to the bottom of the link, instead of to the top, as in all the previous examples. The positions of the slide valve corresponding to those of the link are shown by the full lines *a* and the dotted lines

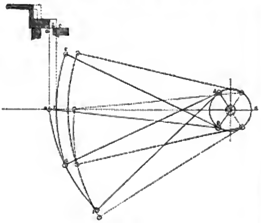

Fig. 154.

c at the top of the diagram. It will be seen by this figure that by the act of placing it in mid gear, the point at which the centre line of the link cuts that of the valve motion is drawn towards the driving axle, instead of being pushed from it, as was the case at this end of the stroke with the other arrangement of eccentric rods; and the lead therefore is decreased, instead of being increased, as the link approaches mid gear. With the exaggerated proportions represented in the diagram under consideration, the valve, when the link is in mid gear, not only has no lead, but covers the port by the amount *b c*.

It is so evident that the decrease of lead arising from this arrangement of the eccentric rods is, like the increase of lead caused by that previously described, subject to modification by variations in the proportions of the eccentric rods, link, &c., that it will be unnecessary to illustrate this by special diagrams.

We must now return to the inquiry which has led to the foregoing remarks,—viz., What is the proper position for the eccentrics? and, assisted by the information given by the preceding figures, it is easy to place these so that they give the required amount of lead to the valve, either when the link is in full gear, or, which we consider the better plan, when it is in half gear. If this latter mode of proceeding is adopted, the lead, instead of being excessively increased when the link is in mid gear, is only slightly so varied, the remainder of the total variation producing a decrease of the lead when the link is in full gear. When the amount of lead fixed upon is that which the valve is to have when the link is in full gear, the eccentrics are to be set, as previously stated, with a linear advance equal to the lap of the valve added to the lead. If, however, the valve is to have this amount of lead when the link is in half gear, the linear advance just mentioned must be diminished or increased (according to the arrangement of eccentric rods employed) by the amount which the lead would be varied by the movement of the link from full to half gear. The extent of this variation may be determined by the aid of a diagram similar to Fig. 151, except that the link must be drawn in half, instead of full gear.

Having shown how the positions of the eccentrics may be fixed upon, we can now proceed to investigate the nature of the movements which they impart to the link.

Motion of the Link derived from Two Eccentrics.—If we suppose the ends of the eccentric rods to be coupled to a "box" link (such as that shown by Fig. 145), and if we further imagine it to be possible, by any arrangement of the centre of suspension of the link, to maintain the centre of the knuckle of one of the eccentric rods coincident with that of the block during the whole stroke when in full gear, it is evident that the motion communicated to the valve will be in every respect identical with that derived from a single eccentric. The motion given to the link by the eccentric not in gear will consist merely of an oscillation round the centre of the block, and can therefore exert no disturbing influence whatever upon the motion of the valve. If, however, as is always practically the case, the centre of the knuckle of the eccentric rod which is in gear does *not* coincide with that of the block throughout the stroke, but is sometimes above and sometimes below that centre, then, whenever such vertical movement occurs, the valve will have its motion modified by the action of the second eccentric; and this modification will be either an acceleration or retardation, according to the relative movements of the two eccentrics at the time of its occurrence. The amount of this variation in the movement of the valve will be increased by an increase in the vertical motion just mentioned, and it will also be influenced by the proportion which the length of the link bears to the throw of the eccentrics.

When treating upon the action of the slide valve, the nature of its motion was explained by means of a diagram showing the various positions which it assumed at different points in the double stroke of the piston or, what is the same thing, during a revolution of the crank shaft. In the same manner the nature of the motion of the expansion link may be best explained by diagrams showing the different positions which it assumes during a revolution of the engine. For the purpose of example it will be convenient that we should determine upon some normal link motion having certain known proportions, and in the following diagrams illustrating the motion of the shifting link, the various parts will (except when a change is specially indicated) be drawn to the following dimensions:—

Throw of eccentrics, 2¼ inches.
Length of eccentric rods, 4 feet 6 inches.
Lap of valve, 1 inch.
Lead in full gear, ¼ inch.
Distance between end centres of expansion link, 12 inches; the point to which the lifting link is attached being situated midway between the end centres and on the centre line of the link.
Centres of lifting links, 15 inches.
Length of lifting arm, from centre of weigh-bar to centre of attachment of lifting link, 14 inches.
Width of ports, &c.,—steam, 1¼ inches; exhaust, 3 inches; bars, 1 inch.

We will commence by investigating the movements which take place when the link is in mid gear, and explain the mode of laying down the successive positions of the link by reference to the diagram, Diagram Plate III., Fig. 1.[*]

Draw the centre line A B, and let any convenient point, C, upon it represent the centre of the crank shaft; through this point draw the line D E perpendicular to A B, and from the same point as a centre describe two circles, the one, B D E, with the length of the crank, and the other with the throw of the eccentrics as a radius. On this latter circle mark off, in the manner already explained, the positions which the centres of the eccentrics should occupy when the crank is in the position C B, or when

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[*] For laying down the positions of the expansion link in the abovementioned and similar diagrams, it is adviasble to employ a templet of one of the forms shown in Fig. 155. The edge C C is formed to the curve of the centre line of the link, and has marked upon it five lines,—one, A, showing the centre of the link, or point of attachment of the lifting link; two others, B B, the points to which the eccentric rods are connected; and the remaining two, C C, the ends of the link. When the studs to which the eccentric rods and lifting link are attached are not upon the centre line of the link, the back of the templet should be made as shown at D or E, so that their position may be marked thereon.

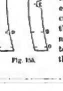

Fig. 155.

the piston is at the commencement of the stroke from the crank shaft. These positions of the eccentrics and crank may be each numbered 1. Next divide both of the circles into any convenient number of equal parts (twelve being the number of divisions adopted in the diagram under consideration), the points of division commencing, in the case of the crank circle, from that already marked 1; whilst in the case of the smaller circle two independent sets of divisions must be made, one series commencing from the centre of the fore-gear eccentric already laid down, and the other from that of the back-gear eccentric. All these sets of divisions are to be numbered in the same order, commencing from the points already marked 1. In the figure, the numbers referring to the positions of the back-gear eccentric are distinguished by the addition of the letter *b*—thus, 1 *b*, 2 *b*, &c.

With the length of the eccentric rods as a radius, and the points 1 and 1 *b* on the small circle as centres, describe the arcs *a* and *c* respectively. Apply to these arcs a templet similar to that shown on Fig. 155, shifting it until the points B B on the templet correspond with the arcs *a* and *c*, while the centre mark, A, agrees with the centre line, A B, on the diagram, and then draw the line *d e*, representing the position of the link at the commencement of the stroke of the piston from the crank. In like manner lay down the position of the link at the commencement of the stroke towards the crank, using in this case the points 7 and 7 *b* as centres for the arcs *f* and *g* respectively. Next, from the points F and G, where the centre lines of the link just drawn intersect the centre line of the valve motion, describe, with the length of the lifting links as a radius, arcs which will intersect each other at H, and from this point, with the same radius, describe the arc F G.

We may now lay down the other positions assumed by the link, using the points 2 and 2 *b*, 3 and 3 *b*, &c., consecutively, and proceeding in the same manner as when obtaining those already drawn, except that the mark on the templet denoting the point of attachment of the lifting link, instead of being made to agree with the centre line of the motion, must be so placed that it coincides with the arc F G, which arc represents the path of the end of the lifting link when the motion is in mid gear. Each of the centre lines of the expansion link thus drawn should be numbered the same as the positions of the eccentrics to which it corresponds; and when the series is completed, a curved line should be drawn through each of the sets of points marking the various positions of the joints of the eccentric rods, and of the ends of the link, and the paths of these parts will thus be indicated. With the box link used in the motion which we have taken as our example, these curves have a shape somewhat resembling an elongated figure eight laid upon its side; and it will be observed that those described by the ends of the link are more open than those described by the points of the eccentric rods, this being owing to the vertical motion of the link induced, at all parts of it except the centre, by its assuming an oblique position, being greater at the former than at the latter points.

The diagram thus completed gives the fullest information respecting the movements of the link. Tracing these out, we find it at the commencement of the stroke of the piston from the crank shaft, vertical; but as the crank shaft revolves, the link assumes an oblique position, this obliquity, which is caused by the two ends of the link being moved, under the influence of the eccentrics, in opposite directions, increasing until position 4 is reached, by which time the crank has become vertical. From this point the obliquity gradually decreases until, when the crank shaft has completed a half revolution, the link has again become upright. After this it again heels over, but in the opposite direction, attaining its greatest obliquity when the crank is near the point 10, or again at right angles to the centre line of the motion; and from this point it gradually assumes the position which it occupied at the commencement of our imaginary revolution of the crank shaft.

Next, as to the extent and nature of the motion given to any particular point in the link. It will be apparent from the diagram that the horizontal movement varies from the centre, where it is least, to the extremities, where it is greatest, and that we can, by moving the link vertically, impart to the block, and consequently to the valve, a greater or less amount of travel, as may be desired. As, moreover, it has been shown in the last chapter that if the lap and lead remain constant, a decrease in the travel is productive of an earlier suppression of the steam, it will be seen that we can, by the aid of the link, easily vary the expansion within certain limits. It is true that, with the "shifting-link" motion, the lead does not remain exactly constant; but it will be remembered that the case of lap remaining unaltered while the lead was increased with the decrease of travel, was specially investigated in the last chapter, and that it was then shown that this increase of lead, unless excessive, was not productive of any ill effects, but that it merely hastened the cut-off and pre-admission of the steam.

The diagram now under consideration gives us the opportunity of investigating the nature of the motion derivable from the centre of the link; and first, it is to be observed that the extreme points of its travel are those which correspond to the extreme points of the stroke of the piston; and we therefore see that the horizontal movement of this part of the link is only equal to twice the travel added to twice the lead,—the lead here referred to being that given to the valve when the link is in mid gear, and not that with which the eccentrics are set. In Fig. 156 the motion of the centre of the link is contrasted with that derivable from an eccentric having equal travel, and having its centre in a line with that of the crank, but on the opposite side of the crank shaft. In this diagram A B is a part of Fig. 1, Diagram Plate III. showing the movements of the central portion of the link; and of the two circles beneath it, the inner one, by its divisions, shows twelve of the positions assumed by the eccentric above mentioned during a revolution of the crank shaft—the divisions of the outer circle denoting in the same way the corresponding positions of the crank pin. The positions of the crank, eccentric, and link, which agree with each other, are numbered alike; and we can thus see, by means of the lines joining the upper and lower portions of the figure, that the motion of the centre of the link is *practically* identical with that which would be given by an eccentric having the proportions, and set in the manner, just described. We have stated that *practically* the positions of the valve given by the link and the above-mentioned eccentric are the same. In reality, however, they are not *precisely* identical; and a little further investigation of Fig. 156 will show the reason of this. Let us take the points 4 and 10 for examples. If the motion of the valve was derived from the eccentric, its position would be the same at each of these points in the revolution of the crank shaft, and, as shown in the figure, it would agree with that given by the intersection of the centre lines of the links Nos. 4 and 10. Owing, however, to the action of the lifting link, the centre of the expansion link is at these points raised above the centre line of the motion, and (the centre of the block continuing on that line) the block is consequently brought in gear with a portion of the expansion link below its centre, and it therefore partakes to a small extent of its back-gear motion. The position of the valve according with position 4 of the link is thus nearer B than it would be if driven by the eccentric; and that agreeing to position 10 of the link is correspondingly nearer A. This irregularity is manifestly greatest at the middle of the travel of the valve, and is capable of being diminished or increased by increasing or diminishing the length of the lifting link. When the point of attachment of the lifting link is not on the centre line of the expansion link, the difference between the motion derived from the centre of the link and that due to an eccentric of equal travel is much greater than that shown by Fig. 156, the vertical movement of the link being under such circumstances considerably increased. This will be apparent when the action of other forms of link is investigated. Before, however, we can determine the nature of the motion imparted to the valve by the other parts of the link, it is necessary that we should arrange the reversing gear by which the link is raised or lowered; and we shall therefore proceed to show the manner in which this is to be done.

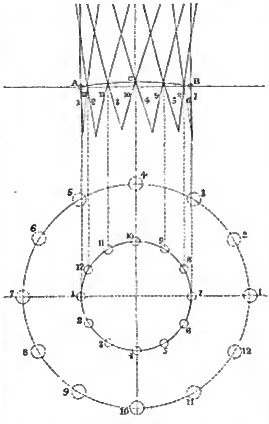

Fig. 156.

Position of the Weigh-bar.—The following mode[*] of obtaining the position of the centre of the weigh-bar is one which has been frequently adopted; and although the position thus fixed upon is not in all cases the best one, yet the method shall be described, as in pointing out the errors sometimes induced by its use, and the manner in which they may be corrected, the influence of the position of the weigh-bar on the motion imparted to the valve can perhaps be more readily explained than in any other way.

In Fig. 157 let *a* and *b* be the positions occupied by the centre of the link, when in mid gear, at the commencements of the strokes of the piston from and towards the crank shaft, and *e* and *f* the positions of the same point when the link is in full gear forward,—these latter being obtained by lowering the link until the centre of the joint of the fore-gear eccentric rod corresponds, as shown at *c* and *d*, with the centre line of the motion. From the points *a b* and *e f* as centres, and with the length of the lifting link as a radius, describe arcs intersecting at *g* and *h*,—these

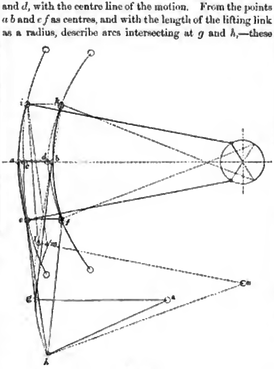

Fig. 157.

points will be the positions of the end centre of the reversing arm for mid and fore gear respectively. With the length of the reversing arm as a radius, and the points

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[*] This method is the one described by Mr. D. K. Clark, in his work on *Railway Machinery.*

g and *h* as centres, describe arcs cutting each other at A, which point is to be the centre of the weighbar, according to this mode of determining it.

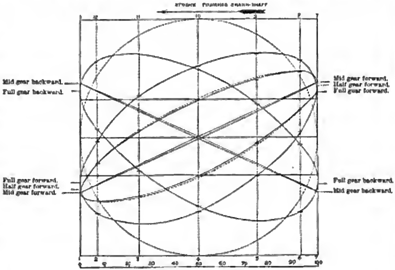

Fig. 158.

Apparently the object here sought to be obtained is that of so placing the weigh shaft that the centres of the joints of the eccentric rods may, when the link is in full gear, be found situated on the centre line of the motion at the commencement of each stroke of the piston, and that, when in mid gear, the centre of suspension may be similarly placed. If, however, we follow out the matter further, we shall find that, with any length of reversing arm practically usable, such conditions can be complied with for one of the full gears only. Thus, in the diagram, let *i* and *k* represent the positions of the centre of suspension of the link at the commencement of each of the strokes of piston when in full back gear, the link being elevated so that the centre of the joint of the back-gear eccentric rod coincides with the centre line of the valve motion. Then, by describing arcs from the centres *i* and *k* with the length of the lifting links as a radius, we obtain the point *l*, which is the position for the extremity of the lifting arm for full back gear. Next, from the centre A, with the length of the reversing arm as a radius, draw the arc *h g*, and continue it upwards, when it will be seen that it will not pass through the point *l*; and if by the ordinary method of practical geometry we determine the centre B, from which an arc could be described that would pass through the three points *l*, *g*, and *h*, we shall find that the radius required will be equal to the length of the eccentric rods, and therefore far greater than is applicable in practice.

Under these circumstances, it is evident that we may place the centre A so that the end of the reversing arm may be able to coincide with any two of the points *l*, *g*, or *h*, or so that it may pass at an equal distance from each of them. As yet, however, it must be remembered that none of these positions of the weigh-bar have been proved to be the best; and to determine whether this is or is not the case it is necessary that we should recur to Diagram Plate III., Fig. 1, and find what would be the motion imparted by the link to the valve under the influence of different arrangements of reversing gear.

In the first instance, the weigh-bar shall be placed so that the end centre of the reversing arm passes through the points *g* and *h*, determined in the manner explained by Fig. 157, the whole of the supposed error caused by the short length of the reversing arm occurring when the link is in full back gear, at which time the end of the reversing arm, instead of occupying the point *l*, will be situated at *m*. The arcs traversed by the centre of suspension of the link in the different gears, being thus obtainable, the positions assumed by the link itself in both half and full gear forward, and full gear backward, may be laid down in the same manner as those corresponding to mid gear, previously delineated; and by the combined aid of these diagrams, the motion curves shown by Fig. 158 may be obtained. The positions taken by the link in full and half gear forward, and full gear backward, are shown on Diagram Plate III., by Figs. 2, 3, and 4.

The diagram of motion curves, Fig. 158, is similarly constructed to that shown on Diagram Plate II., Fig. 2, the circle representing the path of the crank pin, and the datum lines being laid down in the same manner; but instead of the positions of the valve, corresponding to the various points in the revolution of the crank shaft, being obtained by projection from the divisional points of a circle showing the path of the centre of the eccentric, they are derived from the figures on Diagram Plate III., as follows:—To construct the curve corresponding to full gear forward, bisect the distance between the points where the two centre lines 1 and 7 (representing the positions of the link at the commencement of each stroke of the piston), Fig. 3, Diagram Plate III., cut the centre line of motion, and from the point of bisection measure the distance to each of the other points at which those centre lines of the link, which have been laid down, intersect the centre line of motion. Transfer each of the distances so obtained to the correspondingly numbered vertical line in Fig. 158, measuring from the central horizontal datum line, and a series of points on the first-mentioned lines will thus be obtained through which the required curve can be drawn. The curves for mid gear, half gear forward, and full gear backward can of course be similarly described, the measurements being taken in these instances from Figs. 1, 2, and 4, Diagram Plate III., respectively.

Next, let us see what are the results obtained by the application of the process which we have just described to the arrangement of valve gear shown by Figs. 1, 2, 3, and 4, Diagram Plate III., or, as we shall call it, Link No. 1. If the action of the connecting rod is not considered, we find by measurement from Fig. 158, that the points of cut off and release are situated as follows, the positions being expressed in per centages of the stroke measured from its commencement:—

	Stroke towards Crank.	Stroke from Crank.		Stroke towards Crank.	Stroke from Crank.
Full gear forward; steam cut off at	77	76	Exhaust opens at	92¾	91¼
Half gear " "	49	50	" "	79	80
Full gear backward; " "	73¾	74¼	" "	91	90¾

The periods of admission, &c., above given are, as will be seen, very nearly equal for both strokes of the piston, the differences being practically of no importance; if, however, the influence of the connecting rod is allowed for, the result becomes different, as the following table, in which this allowance has been approximately made, will show:—

	Stroke towards Crank.	Stroke from Crank.		Stroke towards Crank.	Stroke from Crank.
Full gear forward; steam cut off at	80	73	Exhaust opens at	94	90¼
Half gear " "	53	45¾	" "	81¾	77¼
Full gear backward; " "	76¼	71¼	" "	92¼	89¼

An inspection of the above table shows that in all cases the period of admission is, with this valve gear, greater during the stroke of the piston towards the crank than during the opposite stroke, and the irregularity would of course be increased by the employment of a shorter connecting rod.

Let us now investigate the cause of this irregularity, and ascertain the means by which it can be decreased. The suppression of the steam during the stroke of the piston *towards* the crank is effected by the motion of the valve *from* the crank, and *vice versâ:* in order, therefore, that the period of admission of the steam during the stroke of the piston towards the crank should be decreased, and that during the stroke from the crank increased, the slide valve must, in the first case, have its motion from the crank shaft accelerated, and in the second, have its motion towards the crank shaft delayed. We shall now show how, in some cases, these requisite alterations in the movements of the valve at different parts of the stroke may be effected by shifting the position of the weigh-bar. The positions of the link at the moments when the cut-off of the steam is effected are, in full gear, approximately those numbered 5 and 11 on Fig. 3, Diagram Plate III.; and it is with the points at which these centre lines of the link intersect the centre line of the motion that we have consequently to deal when altering the points at which the suppression takes place. From what has just been stated it will be clear that, in order to equalize the distribution, it is necessary that the points at which the lines 5 and 11 intersect the centre line should both be removed farther from the crank shaft. This required alteration in the positions of the points of intersection may be obtained by lowering centre line No. 5, and raising No. 11; and this lowering and raising may in its turn be accomplished by moving the weigh-bar (and consequently the end of the lifting arm to which the lifting link is attached) farther from the crank shaft, and thus causing the end of the arc *a*, *a*, which is next the crank shaft, to be depressed, and the other correspondingly elevated.

The corrective action above described will no doubt be understood on tracing out the movements of the different parts, but it may perhaps be explained still more clearly by referring to Fig. 159. In this figure, positions No. 5 and 11 of the link are shown by full lines in the same manner as in Diagram Plate III., Fig. 3, the weigh-bar being in the same position as in that diagram; and in juxtaposition with these are shown, by dotted lines, the altered positions which are assumed by the link at the same points in the revolution of the crank shaft, if the centre of suspension is shifted from A to B, or 5 inches farther from the crank shaft, and at the same time slightly raised. The effect of this alteration is, as was just explained, to cause the centre lines of the link Nos. 5 and 11 to intersect the centre line of the motion at a greater distance from the crank shaft than previously, and thus to hasten the suppression of the steam during the stroke of the piston towards the crank shaft, and delay it during the opposite stroke.

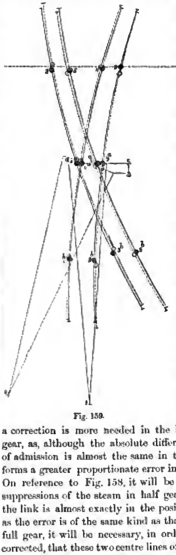

Fig. 159.

So far the working of the motion in full gear only has been considered; the table above given, however, shows that a correction is more needed in the half than in the full gear, as, although the absolute difference in the periods of admission is almost the same in the two cases, yet it forms a greater proportionate error in the latter instance. On reference to Fig. 158, it will be seen that the two suppressions of the steam in half gear are effected when the link is almost exactly in the positions 4 and 10, and, as the error is of the same kind as that which occurred in full gear, it will be necessary, in order that it should be corrected, that these two centre lines of the link should also be made to cut the centre line of the motion at a greater distance from the crank shaft than they now do.

To effect this, No. 4 would require to be lowered and No. 10 raised, and (as these lines do not cross before reaching the centre of the link to which the lifting link is attached), to do this, the centre of the weigh-bar would have to be moved nearer the crank shaft, or the alteration in its position would require to be made in the opposite direction to that necessary in full gear. Owing, however, to the points at which the centre lines 4 and 10 intersect the arc *a*, *a* being so close together, a movement of the centre of suspension, unless of very great extent, would not produce any appreciable elevation or depression of these lines, and consequently would not effect the desired alterations in the position of the points at which the steam is cut off. In fact, in the present instance, this method of correcting the unequal cut-off would not be practically applicable.

In full gear backward the positions of the link at the times when the cut-offs take place are, as will be seen from Fig. 158, nearly those marked 3 and 9, and an investigation similar to the foregoing, applied in this case, will show that the correction required is a depression of the former and elevation of the latter centre line, and consequently (as will be seen by Diagram Plate III., Fig. 4) a movement of the centre of suspension nearer the crank shaft.

Thus far it seems desirable that the arc described by the end of the lifting arm should be less vertical (if such a term be allowable) than shown in Diagram Plate III., Fig. 1, or, in other words, the position of the weigh-bar should be altered so that the upper part of the arc would be situated farther from, and the lower part nearer to, the crank shaft than is there shown. In laying out a correct valve gear, however, there are other points which require attention besides the equal suppression of the steam; and before deciding whether the new points of suspension are preferable to those first fixed upon, it is necessary that we should determine the effect of the alterations upon some of the other qualities of the valve gear under consideration.

In the first place it must be seen whether the extreme openings given to the ports during the two strokes of the piston differ more or less under the new than they did with the old arrangement. Measuring from the Diagram Plate III., Figs. 2, 3 and 4, we find that the extreme openings of the steam ports are as follows:—

	During Stroke towards Crank.	During Stroke from Crank.
Full gear forward; extreme opening of port,	1 3/32 ins.	1 3/32 ins.
Half gear, " " "	15/16 "	15/16 "
Full gear backward; " "	1 3/32 "	1 1/8 "

These results are such that they do not require any improvement; but an inspection of Diagram Plate III, Figs. 2, 3, and 4, will show that the error in full gear backward, slight as it is, would be decreased by the alteration of the point of suspension above proposed for equalizing the distribution of the steam. In full gear forward the port attains its greatest opening when the link is in the positions 3 and 9, and the alteration in the positions of the link, and consequently in the amount of the openings caused by shifting the centre of the suspension farther from the crank shaft, is shown on Fig. 159, from an inspection of which it will be seen that the opening given by the link when in position 3 will be decreased, and that when in position 9, increased, by the new arrangement. The effect in full gear backward may be traced out in a similar manner.

In half gear it has been stated that an alteration in the position of the point of suspension would, in this particular case, be almost inoperative in correcting the unequal distribution of the steam; and it is as well that this should be so, as the alteration, if made, would mar the present equal openings of the ports. Unless, however, the point of suspension was shifted a considerable distance, the alteration effected in the amount of opening given to the ports would not be very great, as the link is (as shown by centre lines Nos. 2 and 8, Diagram Plate III., Fig. 2) so nearly upright at the times when the extreme openings occur, that the horizontal movement of the point of intersection of the centre line of the link with the centre line of motion would be exceedingly slight compared with the vertical movement of the link necessary to produce it.

We now come to another point which, although not affecting the theoretical action of the valve gear, is yet of considerable practical importance—viz., the slip of the block in the link. In the arrangement of valve gear which we are now considering, it will be remembered that the block itself has no vertical movement whatever, and that, consequently, any vertical movement imparted to the link must produce, as a necessary result, a sliding movement between the link and the block. In practice the wear of the block will, other circumstances being equal, be in exact—or almost exact—proportion to the amount of this slip, and it is therefore advisable to reduce the latter to the smallest extent compatible with an efficient distribution of the steam. In Diagram Plate III., Figs. 1, 2, 3, and 4, the slip of the block corresponding to each gear of this particular valve gear may be ascertained by measuring from the end of the link, in each of its positions, to the point in the centre line of the link at which it is intersected by the centre line of the motion; the difference between the greatest and least of the dimensions so obtained being the slip required. The results tabulated are as follow:—

Full gear forward; slip of block, . . . = 11/16 ins.
Half gear " " . . . = 3/32 "
Full gear backward; " . . . = 5/16 "

If the position of the weigh-bar was altered in the manner suggested by the above investigation of the distribution of the steam, the amount of the slip of the block in the different gears would be materially increased, the increase in full gear forward being approximately equal to double the distance *c d* on Fig. 159. In fact, the result is such as to show that, in the case of a box link, having central attachments for the lifting link and eccentric rods, the position of the centre of suspension cannot be varied horizontally to a sufficient extent to correct the irregularity in the distribution of the steam, without producing an amount of slip between the block and the link which would be highly objectionable in practice. In the case of this particular valve gear, therefore, the mode of fixing the position of the weigh-bar, explained by the aid of Fig. 157, may be generally adopted, as the slight inequalities which it produces in the distribution of the steam are lesser evils than the increased slip of the block which would be produced by correcting them.

The reason why the box link with central attachments cannot with advantage have its distribution corrected by the horizontal movement of the point of suspension is (as has no doubt been apparent during the preceding investigation), that the centre lines of the link, when in the positions which it assumes at the times that the steam is cut off, intersect the arc described by the point to which the lifting link is attached, at points so close together that, in order to produce any practically appreciable vertical movement of the link at those particular times, the inclination, so to speak, of the arc must be very considerable; and this excessive inclination of the arc produces a great vertical movement of the link at the points where such motion is not required. Thus, referring to Fig. 159, the necessary depression of the link, represented by the distance between the full and dotted circles 5 *a*, would be accompanied by the occurrence at another point in the revolution of the crank shaft by the unnecessary and detrimental depression *c d*.

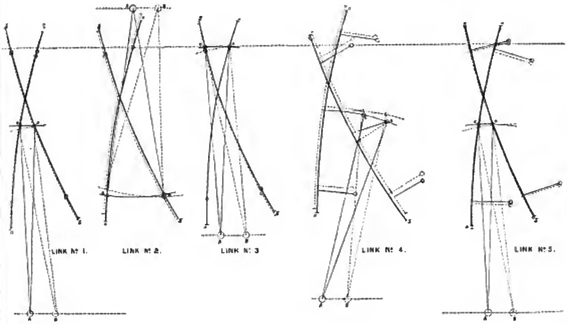

Fig. 160.

We may now see how the other descriptions of link stand in this respect. Fig. 160 shows the effect produced upon five different kinds of links, when in full gear, by a given horizontal movement of the point of suspension, the full line showing the positions of the links when the points of suspension is at A, and the dotted lines representing the altered position when the point of suspension is shifted to B, which is nearer the crank shaft.

The links represented are—No. 1, Box link with central attachments for both eccentric rods and lifting link; No. 2, Link with central attachments for the eccentric rods, and with the lifting link connected to the same point as the back gear rod; No. 3, Link with central attachments for eccentric rods and lifting link connected to same point as fore gear rod; No. 4, Link with both eccentric rods and lifting link attached to knuckles at the back of the link; and No. 5, Link with the eccentric rods coupled to knuckles, as in the last example, but with the lifting link attached to a point on the centre line of the link. In links Nos. 1, 4, and 5 the point of attachment of the lifting link is midway between those of the eccentric rods. As far as the distribution of the steam is concerned, the effects produced on the various examples shown in Fig. 160 by the alteration in the position of the point of suspension are as follows:—

PERIODS OF ADMISSION OF THE STEAM.

	During stroke from Crank Shaft.	During stroke towards Crank Shaft.
Link No. 1, . .	Increased slightly.	Increased.
,, No. 2, . .	,, ,,	,,
,, No. 3, . .	Increased.	Not altered.
,, No. 4, . .	Decreased.	Decreased slightly.
,, No. 5, . .	Not altered.	Increased.

The centre lines represented in Fig. 160 are approximately those which correspond to the positions of the links at the moments when the steam is cut off, and therefore, as has been explained above, the distances C, D between the points at which these centre lines intersect the arcs described by the end of the lifting link represent pretty accurately the relative power which the position of the point of suspension is practically capable of exerting upon the distribution of the steam. Judging by this standard, Link No. 2 possessed a great advantage over the other varieties, and, indeed, with the proportions chosen for our example, the vertical movement given to the link when in the position which it occupies at the moments when the steam is cut off, is very nearly the extreme vertical movement caused at any time by the alteration of the point of suspension. In order that the effect of the position of the weigh-bar may be investigated more fully, we have given, on Diagram Plates III. and IV., a series of diagrams showing the various positions assumed during a revolution of the crank shaft by each of the links represented in Fig. 160, in half gear forward; and have also added similar diagrams of several other forms of shifting link, of which we shall give particulars presently, in half gear. The peculiarities of some of these various forms of link we shall now proceed to consider in order.

Link No. 2.—In Link No. 2 (of which the various positions in full and half gear forward are shown by Fig. 6 and 7, Diagram Plate III.) the lifting link is, as we have already stated, coupled to the same centre as the back gear eccentric rod, which centre, as well as that to which the fore gear rod is connected, is situated upon the centre line of the link. The lifting link has also been made longer than that of Link No. 1, the length between centres being increased to 18 inches. It will be seen that the "loop" described by the point of attachment of the fore gear eccentric rod is much more open in this case than in that of Link No. 1, whilst the path of the joint of the back gear eccentric rod is, of course, reduced to an arc of a circle, having the end of the reversing arm for its centre, and the length of the lifting link for its radius; and it will be seen also that the positions Nos. 5 and 11 of the link (which, with the proportions of valve under consideration, have been shown to be approximately those which the link assumes when in full gear forward at the time that the steam is cut off) intersect this arc very close to its extremities. The effect of this will be seen presently. The position of the weigh-bar has in the diagrams just referred to been determined in the same manner as that for Link No. 1, and the distribution of the steam effected by this link, when so arranged, is shown by the full lines in Fig. 161. The results obtained from this figure are given in the following table, in which the positions of the points of suppression and exhaust are expressed in percentages of the stroke, measured from its commencement:—

	Full Gear Forward.		Half Gear Forward.	
	Stroke towards Crank.	Stroke from Crank.	Stroke towards Crank.	Stroke from Crank.
Steam cut off at	74¾	78½	49½	55¼
———— corrected for the action of a six-length connecting rod,	77¼	75½	53½	51¼
Exhaust opens at	91¾	92½	78¾	79¼
———— corrected for the action of a six-length connecting rod, .	93	91¼	81½	74¾
Extreme opening, ...	1$\frac{1}{16}$ in. full	1¼ in.	$\frac{13}{16}$ in.	$\frac{11}{16}$ in.
Slip of Block in Link, . . .	¼ in.		$\frac{11}{16}$ in.	

The above table shows that the distribution of the steam effected by this link, when arranged as above described, and used in conjunction with a connecting rod six times the length of the crank, is very good, and that the periods of admission, &c., are more equal than was the case with Link No. 1 under similar circumstances. The openings of the ports are, however, more unequal than with that link, and the slip of the block is greater. An inspection of Figs 6 and 7, Diagram Plate III., will show that the correction of the slight irregularities in the distribution of the steam, and that of the inequality of the port openings, given by the link in full gear, would require movements of the weigh-bar in opposite directions; and an attempt to equalize the openings would result, therefore, in an increased difference between the periods of admission.

Thus the reduction of the opening of the one steam port during the stroke from the crank shaft, and an increase in the opening of the other port during the opposite stroke, would require the link to be raised when in position 3, and lowered when in that marked 9; and this would involve a movement of the point of suspenion *further from* the crank shaft. On the other hand, the reduction of the period of admission during the stroke towards the crank shaft, and its increase during the other stroke, would necessitate the lowering of the link when near position 5, and its elevation when in position 11; and to do this the point of suspension would have to be moved *nearer to* the crank shaft. If, therefore, an alteration in the position of the weigh-bar alone be employed to adjust the performance of the link when in full gear, it will be necessary to balance the advantages of equal admissions against those of equal openings for the ports, and to proceed according to the decision arrived at.

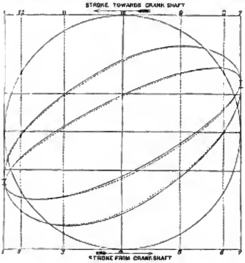

Fig. 161.—Motion Curves for Links Nos 2 and 2 a.

The various considerations which should affect this decision will be discussed in a future part of this work, after the various means of adjusting valve motion have been explained; but we may mention here that it should be remarked that the alteration in the position in the point of suspension necessary to equalize the periods of admission would increase the slip of the block, whilst that required to equalize the openings would slightly reduce it. The positions of the points at which the centre lines Nos. 3, 9, 5, and 11 intersect the arc described by the end of the lifting link show also that the two first-mentioned centre lines, which govern the extreme openings of the ports, would be less affected by any given movement of the point of suspension than the two latter, which approximately regulate the points at which the steam is cut off when the valve motion is in full gear.

It generally happens, however, that a link like No. 2—having the lifting link attached to the same point as the back gear eccentric rod—is made as shown in Fig. 148; and it is thus rendered incapable of being worked in full gear, according to the meaning which has hitherto been attached to that term, or more correctly, it cannot be placed so that the centre of the block shall coincide with either of the centres of attachment of the eccentric rods.

Under these circumstances the action of this valve motion in intermediate gears is evidently of much greater importance than that in what has been called full gear; and it is therefore necessary that before leaving this link we should direct some attention to Fig 7, Diagram Plate III., which shows its working in half gear forward.

The table already given shows that when the link is in this gear there is (the connecting rod being equal to six times the length of the crank) a slight excess in the period of admission during the stroke towards the crank shaft, and an excess in the opening of the port during the opposite stroke. The first of these errors might be corrected by lowering the link when in position 4, and raising it when in position 10; and the openings during the two strokes might be equalized by raising the link when in position 2, and lowering it when in position 8; and an inspection of Fig. 7, Diagram Plate III., will show that all these movements might be effected by shifting the point of suspension nearer the crank shaft. The dotted lines in Fig. 161 show the alterations in the distribution of the steam—both in full and half gear—effected by moving the point of support of the link 2 inches nearer the crank shaft; and the results of the alteration are given in the general table below, from which it will be seen that the periods of admission and the openings have been rendered more nearly equal, but the slip of the block has been increased.

From the above results we see that when the link is in such a gear that the points of intersection of each of the two pairs of centre lines which represent the positions of the links at the times that the suppression of the steam and the extreme openings of the ports occur, are either both above or both below the arc described by the end of the lifting link, the movement of the point of suspension necessary to correct the inequality in the periods of admission will be in an opposite direction to that required to correct the inequality in the amounts of opening of the ports; and if, on the other hand, one of the pairs of lines have their point of intersection above, and the other pair below the arc described by the end of the lifting link, the point of suspension will have to be shifted in the same direction, to equalize both the periods of admission and the openings of the ports.

We have dealt with the peculiarities of Links Nos. 1 and 2 at considerable length, but it will not be necessary that we should give the various other forms of link such detailed consideration. The principles which we have explained apply equally to all; and by carefully comparing the diagrams on Diagram Plates III. and IV., with the distribution of the steam effected by the links, as given in the following table, all the peculiar properties of each variety of link can be readily traced out without special explanation on our part. It will, however, be convenient for the purpose of reference if we give here a short description of each of the varieties of link referred to in the table.

Link No. 1—Figs. 1, 2, 3, and 4, Diagram Plate III.—Box link with central attachments for the eccentric rods with lifting link. Parts of gear of the dimensions given on page 164, and weigh-bar set by method explained on page 166. Distribution of steam shown by Fig. 158.

Link No. 1 *a*—Fig. 5, Diagram Plate III.—Same as link No. 1, but with the weigh-bar placed above the centre line of the valve motion and the lifting link reduced in length to 6 inches. Distribution of steam in half gear forward shown by dotted line on Fig. 158.

Link No. 2—Figs. 6 and 7, Diagram Plate III.—Link with lifting link attached to same centre as back gear eccentric rods. Dimensions of parts the same as given on page 164, except lifting link, which is 18 inches long. Weigh-bar set by method explained on page 166. Distribution of steam shown by full lines on Fig. 161.

Link No. 2 *a*—Same as link No. 2, but with weigh-bar moved 2 inches nearer the crank shaft. Distribution of steam shown by dotted lines on Fig. 161.

Link No. 3—Fig 8, Diagram Plate III.—Link with lifting link coupled to the same centre as the fore gear eccentric rod. Dimensions of parts the same as given on page 164, except the lifting link, which is 18 inches long. Weigh-bar set by method explained on page 166. Distribution of steam shown by full lines on Fig. 162.

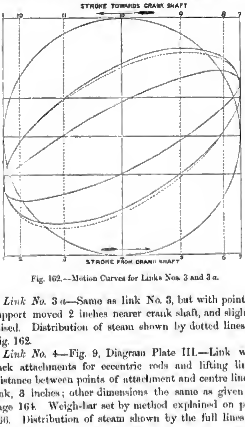

Fig. 162.—Motion Curves for Links Nos. 3 and 3 a.

Link No. 3 *a*—Same as link No. 3, but with point of support moved 2 inches nearer crank shaft, and slightly raised. Distribution of steam shown by dotted lines on Fig. 162.

Link No. 4—Fig. 9, Diagram Plate III.—Link with back attachments for eccentric rods and lifting links. Distance between points of attachment and centre line of link, 3 inches; other dimensions the same as given on page 164. Weigh-bar set by method explained on page 166. Distribution of steam shown by the full lines on Fig. 163.

Link No. 4 *a*—Fig. 10, Diagram Plate IV.—Same as Link No. 4, but with the weigh-bar moved 6 inches farther from crank shaft, and raised $2\frac{3}{8}$ inches. Distribution of steam in half gear forward shown by the dotted lines on Fig. 163.

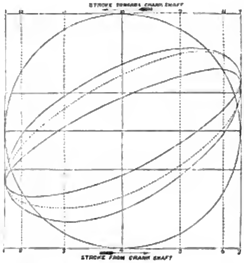

Fig. 163.—Motion Curves for Links Nos. 4 and 4*a*.

Link No. 5—Fig. 11, Diagram Plate IV.—Link with eccentric rods attached to points 3 inches behind centre line of link, but with the lifting link coupled on the centre line at the middle of its length. Dimensions of parts same as given on page 164. Weigh-bar set by method explained on page 166. Distribution of steam shown by Fig. 164.

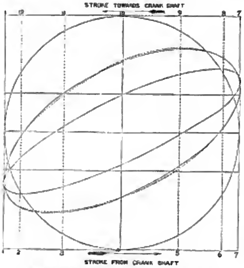

Fig. 164.—Motion Curves for Links Nos. 5 and 5*a*.

Link No. 5*a*—Same as link No. 5, but with weigh-bar moved 3 inches farther from crank shaft. Distribution of steam in full gear forward shown by dotted lines on Fig. 164.

Link No. 6—Fig. 12, Diagram Plate IV.—Link with eccentric rods coupled to points 3 inches behind the centre line, and with the lifting link attached to the same centre as the back gear eccentric rod. Dimensions of parts the same as given on page 164, except that the lifting link is 18 inches long. Weigh-bar set by method explained on page 166.

Link No. 7—Fig. 13, Diagram Plate IV.—Link with eccentric rod coupled to points 3 inches behind centre line, and with the lifting link attached to the same centre as the fore gear eccentric rod. Dimensions of parts the same as given on page 164, except that the lifting link is 18 inches long. Weigh-bar set by method explained on page 166.

Link No. 8—Fig. 14, Diagram Plate IV.—Link with eccentric rods coupled to points 3 inches behind the centre line, and with the lifting link attached to a point on the centre line 6 inches below the centre of link. Length of lifting link 18 inches; other parts of the dimensions given on page 164. Weigh-bar set by method explained on page 166.

Link No. 9—Fig. 15, Diagram Plate IV.—Link with eccentric rods attached to points 3 inches behind the centre line, and with the lifting link coupled to a point on the centre line 6 inches above the centre of the link. Lifting link 18 inches long; other parts of the same dimensions as given on page 164. Weigh-bar set by method explained on page 166. Distribution of steam shown by Fig. 165.

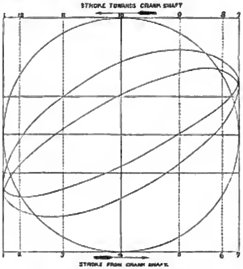

Fig. 165.—Motion Curves for Link No. 9.

Link No. 10—Fig. 16, Diagram Plate IV.—Link with eccentric rods attached to points 3 inches behind the centre line of the link, and with the lifting link coupled to a point on the centre line 9½ inches above the centre, the weigh-bar, which is set by the method explained on page 166, being also above. Lifting link 18 inches long; other parts of the same dimensions as given on page 164.

Link No. 11—Figs. 17 and 18, Diagram Plate IV.—Link with eccentric rods and lifting link coupled to points on the centre line, the point of attachment of the lifting link being 9½ inches below the centre of the link. Lifting link 18 inches long; other parts of the dimensions given on page 164. Weigh-bar set by the direct method explained by Fig. 167. Distribution of steam shown by Fig. 166.

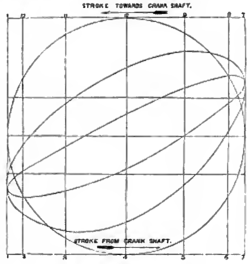

Fig. 166.—Motion Curves for Link No. 11.

Hitherto we have, except in the case of Link No. 11, always supposed the position of the weigh-bar to be first determined by the method explained on page 166, and have then shown how the inequalities of distribution, &c., sometimes incidental to that method, may in many instances be diminished or removed by a system of trial and error. It is, however, quite possible to determine at once that position of the weigh-bar which will give a perfectly equal period of admission during both strokes of the piston; and we shall now explain a method of doing this which is, we believe, new.

In Fig. 167, let A B be the centre line of the valve motion, and C the centre of the crank shaft, and from this latter point as a centre describe two circles, the one with a radius equal to the length of the crank, and the other equal to the throw of the eccentric. Then let the diameter D B of the larger circle represent the stroke of the piston, and from each end of this diameter mark off two periods of admission which are known to be within the range of the valve gear which is to be laid down. In the figure the periods of admission chosen are 40 and 70 per cent., the general proportions of the valve and gear being, with the exception of the length of the lifting link (which is 18 inches), the same as given on page 164. Next, through

TABLE SHOWING THE WORKING OF THE VARIOUS DESCRIPTIONS OF SHIFTING LINKS SHOWN ON DIAGRAM PLATES III. AND IV.

Description of Link and Reference to Figures on Diagram Plates III. and IV.	Gear.	Uncorrected Periods of Admission, &c., in percentages of the Stroke, measured from its commencement.				Periods of Admission, &c., in percentages of the Stroke, measured from its commencement, and corrected for the action of a Connecting Rod six times the length of the Crank.				Extreme Opening of Port.		Lead.		Maximum Slip of the Block in the Link.
		Stroke towards Crank.		Stroke from Crank.		Stroke towards Crank.		Stroke from Crank.		During the Stroke towards the Crank.	During the Stroke from the Crank.	End next Crank.	End furthest from Crank.	
		Steam Cut Off.	Exhaust Opens.	Steam Cut Off.	Exhaust Opens.	Steam Cut Off.	Exhaust Opens.	Steam Cut Off.	Exhaust Opens.					
										Ins.	Ins.	Ins.	Ins.	Ins.
Link No. 1 (Fig. 3),	Full gear forward,	77	92¾	76	91¼	80	91	73	90¼	1 9/32	1 5/16 full	⅛	⅛	1⅛
,, ,, (Fig. 2),	Half gear forward,	49	79	50	80	53½	81⅞	45⅞	77¼	¾	⅞	⅛	3/32	7/32
,, ,, (Fig. 4),	Full gear backward,	73¼	91	74½	90¾	76½	92¼	71¼	89½	1 1/16	1⅛	⅛ full	⅛	9/16
Link No. 1a (Fig. 5),	Half gear forward,	50¾	79½	52	80¾	54½	82½	47¾	77¾	⅞	⅞	⅛	⅛	⅛
Link No. 1 (Fig. 6),	Full gear forward,	74¼	91⅞	78½	92¼	77½	93	75½	91¼	1 5/16	1¼	¼	⅛	1
,, ,, (Fig. 7),	Half gear forward,	49¾	78⅞	55¼	79¼	53¾	81⅞	51¼	76⅞	¾	⅞	⅛	⅛	1 3/16
Link No. 2a, . .	Full gear forward,	73	91	77¾	92	76¼	92¼	74¾	90¾	1 1/16	1⅜	¼	⅛	1 7/16
,, ,, . .	Half gear forward,	48¼	77¼	56	81¼	52½	80¼	51¾	79	⅞	⅞	⅛	⅛	1 5/16
Link No. 3, . .	Full gear forward,	75	91¼	75	91½	78¼	92¾	71⅞	90⅛	1 5/32	1 5/32	¼	⅛	⅛ full
,, ,, (Fig. 8),	Half gear forward,	46¾	76¾	48	77	51	79¾	43⅞	74	¾	¾	⅛	⅛	⅛ full
Link No. 3a, . .	Full gear forward,	73¾	91¼	75¼	91½	77	92¾	72¾	90¼	1 7/32	1⅛	⅛	⅛	⅛
Link No. 4, . .	Full gear forward,	73	91¼	78	92½	76¾	92⅞	75	91¼	1 5/16	1⅜	¼	⅛	2 bare
,, ,, (Fig. 9),	Half gear forward,	43¾	75	54¼	81¼	47¾	78¼	50¼	76	⅞ full	¾ full	⅛	⅛	1 3/16
Link No. 4a (Fig. 10),	Nearly full gear forward, .	65	87	72¼	91	69	89	69	89½	1 3/32	1 3/16	3/16	3/16	2 3/16
Link No. 5, . .	Full gear forward,	76½	92¼	70¾	90¼	79½	93¼	67¼	89	1⅛	1 3/16 b.	⅛	⅛	3/32
,, ,, (Fig. 11),	Half gear forward,	50	79¼	43¾	75	54¾	82	39¾	71⅞	⅞ full	¾	⅛	⅛	3/32
Link No. 5a, . .	Full gear forward,	76¼	91¼	72	91½	79½	92¼	68¾	90	1¼ full	1 3/32 b.	⅛	⅛	⅛
Link No. 6, . .	Full gear forward,	70¾	89½	78	92	74½	91	75	90¾	1½	1 7/16 full	⅛	⅛	2 7/16 full
,, ,, (Fig. 12),	Half gear forward,	42¾	74	57¼	81½	46¾	77¼	53¼	79	⅞	1 3/16	1/16 full	1/16 full	2 1/16
Link No. 7, . .	Full gear forward,	69¼	89½	75½	90¾	73	91	72¼	89¼	1¼ bare	1 3/16 full	⅛	⅛	2 1/16
,, ,, (Fig. 13),	Half gear forward,	39	71	51½	78½	43	74½	47¼	75½	⅞ full	⅞ full	⅛	⅛	2
Link No. 8, . .	Full gear forward,	76½	91½	72½	90	79½	92¾	69	88½	1¼	1¼	⅛ full	⅛ full	1⅛ full
,, ,, (Fig. 14),	Half gear forward,	51½	79¼	47¼	76	55¼	82½	43	73	⅞ bare	⅞ full	⅛	⅛	3/16 full
Link No. 9, . .	Full gear forward,	76	91¾	69½	89¼	79	92¼	66	87¾	1¼	1 1/16	⅛ full	⅛ full	3/32
,, ,, (Fig. 15),	Half gear forward,	49¼	78	42	73½	53¾	80¾	38	70¼	¾	⅞	⅛	⅛	3/32
Link No. 10, . .	Full gear forward,	74½	91	68½	90	77¼	92¼	65	88½	1 3/16	1¼	¼	⅛	1/16 bare
,, ,, (Fig. 16),	Half gear forward,	42	77	39¼	73	46	80	35¼	69¾	⅞	⅞	⅛	⅛	¼
Link No. 11 (Fig. 17),	Full gear forward,	72¾	90¼	79½	92	76½	91¾	76½	90¾	1 5/16	1 3/16	⅛	⅛	1⅜
,, ,, (Fig. 18),	Half gear forward,	36	67¾	44¼	74	40	71	40	70¼	7/16	⅜	1/16	1/16	1

each of the points on the line D B marking the cut-off of the steam, describe with a radius equal to the length of the connecting rod arcs cutting the crank circle, the centres from which these arcs are described being all situated on the centre line of the motion in front of, or on the cylinder side of, the crank shaft. In the figure these arcs are shown cutting the crank circle at the points 2, 3, 4, 6, 7, and 8; and lines drawn from these points to the centre, C, will represent the positions of the crank at the times that the piston arrives at the ends of the various periods of admission. Thus C 8 is the position of the crank when the piston has performed 70 per cent. of its stroke towards the crank shaft, and C 4 its position when it has travelled 70 per cent. in the opposite direction, the engine being supposed to be running forward. In like manner, C 7 and C 3 show the positions of the crank when the piston has performed 40 per cent. of its strokes towards and from the crank shaft respectively. When the engine is running backwards, the positions of the crank after the piston had completed 70 per cent. of its strokes towards and from the crank shaft are respectively shown by C 2 and C 6. The lines C 1 and C 5 of course show the positions of the crank at the end of each stroke of the piston.

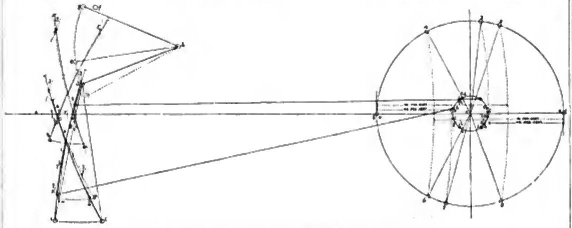

Fig. 167.—Direct Method of determining the Position of the Weigh-bar.

The positions of the crank having been determined, the corresponding positions of the eccentrics must be next marked off on the smaller circle, each position being numbered to agree with the corresponding position of the crank, and the positions of the back gear eccentric being distinguished by affixing some mark, the letter *b* for instance, as has been done in the figure. Next, lay down, in the manner already explained, the position assumed by the link when in mid gear at the end of each stroke of the piston, and mark the points at which these centre lines of link cut the centre line of motion. In Fig. 167 the centre lines of the link just referred to have, for the sake of clearness, been omitted; but the points at which they would intersect the centre line of motion are shown, and are marked 1 and 5, to agree with the corresponding positions of the crank. By bisecting the distance between the points 1 and 5 just mentioned, the datum point F is obtained, this point marking the middle of the traverse of the centre of the link when the latter is in mid gear.

Now, when the centre of the block in the link is situated on the point F, it is evident that the valve projects over both steam ports to the extent of its lap, and that if the block be moved to one side or the other of the point F to the extent of the lap, one port or the other will be just on the point of opening or closing, as the case may be. If, therefore, we mark off from the point F on the centre line of motion two distances, F H, F G, each equal to the lap of the valve, we shall know that whenever the centre line of the link intersects the centre line of motion at either of the points G or H, one steam port or the other is either just about to open or is just closed, according to the direction of the motion of the valve at the time.

The fact just mentioned forms the whole key to the method of setting the weigh-bar which we are now describing. From what we have said it will be evident that all that it is necessary to do to ensure equal admissions of steam during the two strokes of the piston is to so place the weigh-bar that the centre line of the link, in any particular gear, cuts the points G and H respectively at times when the piston has performed equal proportions of each stroke. Thus, if the link is in such gear that the steam is cut off at 70 per cent of the stroke, the centre line of the link should cut the points G and H when the crank is in the positions C 4 and C 8 respectively, the valve being in each case moving towards the centre of its stroke. The mode of fixing the position of the weigh-bar so that the centre line of the link may intersect the points G and H in this manner, we shall now proceed to explain.

Let us first consider that gear in which the cut-off takes place at 70 per cent. of the stroke. From the points 8 and 8 *b* on the smaller circle (which respectively represent the positions of the fore and back gear eccentrics when the crank is in the position C 8) as centres, and with the length of the eccentric rods as a radius, describe the short arcs marked 8 and 8 *b* towards the left of the figure, and in the same manner describe arcs 4 and 4 *b*, from the centres 4 and 4 *b* on the small circle respectively. Now, as we have already said, C 8 represents the position of the crank when the piston has performed 70 per cent. of its stroke towards the crank shaft, and at that time, therefore, the points of attachment of the eccentric rods to the link must be situated somewhere on the arcs 8 and 8 *b* just drawn. But it is also desired that the steam should be cut off at 70 per cent. of the stroke; and whilst, therefore, the centres of attachment of the eccentric rods are situated on the arcs 8 and 8 *b*, the centre line of the link must cut the point H. In order that it may do this, it is only necessary to apply a template of the link to the arcs 8 and 8 *b*, so that the points of attachment of the rods are situated on these arcs, and then to move the template up or down, still keeping the eccentric rod centres on the arcs, until the centre line of the link cuts the position H; and when this required position of the link has been determined, it may be drawn in, as has been done in the figure. In the same way, there should also be laid down the position which should be occupied by the link when the piston has performed 70 per cent. of its stroke from the crank shaft, the arcs 4 and 4 *b* being in this case used in the same manner as the arcs 8 and 8 *b* had previously been. Next should be drawn the positions assumed by the link when cutting off the steam at a shorter percentage of the stroke, and also the positions taken by it when cutting off at some fixed percentage in back gear. In Fig 167 the link is drawn in its positions for cutting off at 70 and 40 per cent. of the stroke in fore gear, and 70 per cent. of the stroke in back gear (the centre lines in the latter instance being dotted), and each centre line of the link is numbered to correspond with the position of the crank to which it corresponds, the arcs passing through the centres of attachment of the eccentric rods to the link being numbered, to agree with the centres of eccentrics from which they were struck.

The link represented in Fig. 167 has the points of attachment of the eccentric rods situated on its centre line; and we will presume, for the sake of example, that it is at first attempted to place the point of attachment of the lifting link also on this centre line, and at the middle of the length of the link. The effect of this arrangement is shown in Fig. 168, in which the centre lines 4 and 8, 3 and 7 of the link are shown in the same manner as in Fig. 167, and the position of the point of attachment of the lifting link corresponding to these two pairs of centre lines are shown at I *i* and K *k* respectively. If, with the two pairs of points last mentioned as centres, and with the length of the lifting link as a radius, we describe two pairs of arcs intersecting each other as shown at *a* and *c*, these two latter points will indicate the positions which should be occupied by the end of the lifting arm, in order that the links should take the positions shown in the figure.

The positions *a* and *c* of the end of the lifting arm are, however, impracticable, and so are those assumed by the lifting link; and it is evident, therefore, that the latter must be attached to some other point of the link if we wish that link to take the positions shown in Fig. 167. When treating of Link No. 1, we remarked upon the small governing power which could be exerted by altering the position of the weigh-bar when the lifting link was attached to the expansion link at the middle of the length of the centre line of the latter, and Fig. 168 shows this fact very strongly. By shifting the point of attachment of the lifting link nearer the end of the expansion link, greater control is obtained; and in Fig. 167 the lifting link is shown coupled to a point 3½ inches below the point of attachment of the back gear eccentric rod, the position of this point on the different centre lines being shown at L *l*, M *m*, and N *n* respectively.

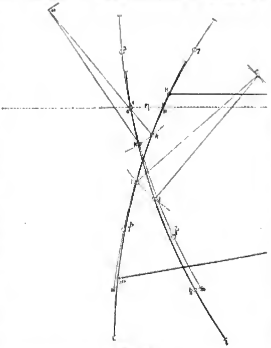

Fig. 168.

From these positions of the point of attachment of the lifting link we may now fix the position of the weigh-bar as follows:—From each of the points L *l*, representing the position of the point of attachment of the lifting link when the steam is being cut off at 70 per cent. of the stroke, describe, with the length of the lifting link as a radius, arcs which will intersect each other at the point *d*, and this point will indicate the position which should be occupied by the end of the lifting arm when the link is in this particular gear. In like manner, by describing arcs from the points M *m* and N *n*, which respectively mark the positions occupied by the point of attachment of lifting link when the steam is being cut off at 40 per cent. of the stroke in fore gear, and 70 per cent. in back gear, the positions *e* and *f* of the end of the lifting arm can be obtained, and all that is then necessary is to describe an arc passing through the points *d*, *e*, and *f*, when the centre of that arc will be the centre of the weigh-bar.

If, however, the length of the lifting arm has been already fixed upon, the position of the centre of the weigh-bar may be determined, as shown in Fig. 167, by describing arcs with this length as a radius, and the points *d* and *e* as centres, the intersection of those arcs at *h* marking the centre of the weigh-bar. It will be seen, however, that in Fig. 167 an arc described from the centre *h*, with the length of the lifting arm as a radius, does not pass through the point *f*, and the positions taken by the end of the lifting arm when the link is in back gear will not be exactly correct, the position when steam is being cut off at 70 per cent. of the stroke being at *g* instead of *f*. The positions of the link given in full lines show the result of the end of the lifting arm being at *g*, whilst those given in dotted lines correspond to the position marked *f*, and it will thus be seen that the disturbance caused by the arm being shifted from *f* to *g* is not very great.

Fig. 169 shows another form of link—one with back attachments for the eccentric rods—laid down in the manner which we have described, and in this instance the lifting link is attached to the same point as the fore gear eccentric rod. The various parts on this figure are numbered and lettered to correspond with Figs. 167 and 168, and no special explanation will therefore be required.

Although the position of the weigh-bar determined in the manner shown by Fig. 167 gives perfectly equal periods of admission during both strokes of the piston, yet with most forms of links the amount of slip which it induces between the link and block is too great to allow it to be exactly adopted in practice. Thus, in the link shown in Figs. 167 and 168, the amount of this slip when the steam is being cut off at 70 per cent. of the stroke in fore gear is 1½ inches, whilst in the case of the link shown in Fig. 169 it is 2⅞ inches. The diagrams, however, show that these amounts of slip could be reduced without greatly affecting the correctness of the distribution; and in laying out a link motion practically, it must always be to some extent left to the judgment of the designer how far this correctness of distribution shall be sacrificed in order to reduce the wear of the link and block.

The distribution produced by the use of the open link, with the eccentric rods connected at points back of the centre line, differs considerably from that given by the box link, on account of the vertical movement of the ends of the eccentric rods, which takes place as the link varies its angle of inclination. When the top of the link is inclined forward, the eccentric rods will be thrown up, and this will bring the upper centre nearer to the shaft, and thereby reduce the inclination of the link, the effect of which will be to diminish the opening of the valve, and to hasten the cut-off on that end, or on the stroke from the crank. When the link is inclined in the opposite direction, the end of the eccentric rods will be thrown down, and owing to the position of the eccentrics at this time, the top one will be again carried nearer to the shaft, but only slightly. This will increase by a small amount the opening of the valve on this end, and delay the cut-off. This action of the horizontal displacement of the ends of the eccentric rods will be found again explained by the aid of a figure, showing the motion of the eccentric rods, in the chapter on the Stationary Link Motion. The results given by a link in which the eccentrics are connected 3 inches back of the centre are given in the following table:—

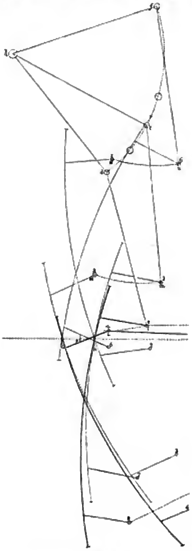

Fig. 169.

	Uncorrected Values.				Corrected for Six-Length Rod.				
	Lead.	Greatest Opening.	Suppression.	Exhaust.	Lead.	Greatest Opening.	Suppression.	Exhaust.	
Stroke towards Crank, .	·28	1·25	·76½	·92	·28	1·25	·79¼	·93¼	Full Gear Forward.
Stroke from Crank, . .	·28	1·18	·71	·89¾	·28	1·18	·67½	·88¼	
Stroke towards Crank, .	·42	·68	·51¾	·79¾	·42	·68	·55½	·82¼	Half Gear Forward.
Stroke from Crank, . .	·42	·65	·43¾	·75	·42	·65	·39	·71¾	

This distribution, it will be seen, is extremely bad; but there are a number of ways in which it can be rectified. Thus, it will be observed that the errors in this case are exactly the reverse of those produced by moving the suspended centre back of the centre line of the link, and this indicates the most effectual mode of correcting the distribution given by the open link—viz., by placing the suspended centre on the same side of the centre line of the link as the eccentric rod joints. As we may see by comparing the results already obtained with the suspended centre so moved with those produced by removing the eccentric rod joints, the former change is more potent than the latter; and it follows, therefore, that to produce a correct distribution, the suspended centre will not require to be moved as far from the centre line as the eccentric rod joints, but to an intermediate position, which can be easily determined by trial. The other means by which an equal distribution may be obtained with an open link having the eccentric rods connected behind, either in fore gear, or in both fore and back gears, will be explained in the succeeding chapter in relation to the mode of connection between the link and valve.

CHAPTER IX.*

INFLUENCE OF THE CONNECTION BETWEEN THE LINK AND VALVE.

THUS far the valve spindle has been supposed to move in fixed guides, and the path of the block, therefore, to be a straight line. It is quite common, however, to convey the motion of the link to the valve stem through the medium of a jointed link suspended from some fixed point. It now remains to be shown in what way this method enables us to modify the motion of the valve by varying the position of the fixed centre from which the valve-rod link is suspended, as well as by alterations in the relative dimensions of the suspending and valve-rod links, and the distance from the ends of the latter at which the former is attached. As the end of the suspending link describes an arc of a circle, it will be seen that the block in the link will describe either a circular or an elliptical arc, according as the point of attachment of the suspending link coincides in position with the centre of the block, or is placed nearer to, or further from, the forward centre of the valve-rod link, or that at which it joins the valve stem. If the valve-rod link is suspended behind the link, the curvature of the path of the block will be less than that of the end of the suspending link; and if the connection is made forward of the link, the curvature will be greater than that of the circular arc of the suspending link. In either case, however, as the arcs are comparatively short, they may be drawn as circular, with radii greater or less than the length of the suspending link, as the case may be. By finding the position of the block at three points in its path, we can by the usual geometrical construction find the centre and length of radius with which the curve should be drawn.

To ascertain the effect of this mode of connection, we have only to draw such curves across the figures in Diagram Plate III., representing the successive positions of the centre line of the link, and from the points where they intersect these lines, with the length of the valve-rod link as a radius, lay off on the centre line of the motion points which will represent the successive positions of the valve, from which we can construct motion curves, as in the examples already given. If the fixed centre of suspension be placed perpendicularly above or below the centre line of the motion, at the central point of its vibration, between the positions at the beginning of each stroke, the modification of the distribution produced by the use of the suspending link will be very simple, and readily understood from Fig. 2, Diagram Plate III., in which the path of the block, supposing the valve rod suspended from above, is shown in dotted lines. As the curve crosses the centre line of the motion in positions 1 and 7, it follows that the lead will remain unaffected; and as the block is brought nearer to the centre of the link at the time of closing of the valve, and still more at the instant of opening of the exhaust, it is evident that these occurrences will be somewhat hastened. On the other hand, as the curve rises above the centre line when in its extreme positions, it follows that the greatest opening of the valve will be increased. Exactly the reverse of these effects will be produced in back gear, since the block would be at a greater distance from the centre of the link in the central positions of the latter, and nearer that point when at the extremities of its motion. Of course, if the fixed centre is placed below the centre line, all these effects will be reversed. The following table gives the comparative results for full and half gear forward of the normal motion Table, and one in which a suspended valve rod is employed, its length being taken as 27 inches, the length of the suspending link 8 inches and the distance from the centre of the block at which the latter is connected, 6 inches:—

TABLE SHOWING EFFECT OF VALVE-ROD LINK SUSPENDED FROM ABOVE.

	UNCORRECTED VALUES.				CORRECTED FOR A SIX-LENGTH ROD.					
	Lead.	Greatest Opening.	Suppression.	Exhaust.	Lead.	Greatest Opening.	Suppression.	Exhaust.		
Stroke towards Crank, .	·25	1·28	·76	·91⅛	·25	1·28	·80	·94	Normal Motion.	Full Gear Forward.
Stroke from Crank, . .	·25	1·35	·77	·92⅞	·25	1·35	·73	·90⅛		
Stroke towards Crank, .	·25	1·32	·75¾	·91¾	·25	1·32	·80	·93¾	Motion with Valve-Rod Link.	
Stroke from Crank, . .	·25	1·40	·76¾	·92¼	·25	1·40	·72¾	·89¾		
Stroke towards Crank, .	·42	...	·49	·79	·42	...	·53¼	·81¾	Normal Motion.	Half Gear Forward.
Stroke from Crank, . .	·42	...	·50	·80	·42	...	·45¾	·77¼		
Stroke towards Crank, .	·42	...	·48¼	·78⅛	·42	...	·52⅛	·81¼	Motion with Valve-Rod Link.	
Stroke from Crank, . .	·42	...	·49¼	·79⅛	·42	...	·45	·82⅛		

The changes produced, therefore, with this arrangement of the parts, are not of any great importance, as they do not affect at all the *equality* of the distribution on the two ends, but only slightly modify the speed with which the valve closes. As the port opening is wider when the valve-rod link is used, and closes earlier, the cut-off must be a little sharper, which is, of course, an advantage as far as it goes, though it only exists in fore gear, the reverse being true for back gear. This, however, is gained at the expense of a greater slip of the block in the link, with its attendant disadvantages.

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* By MR. FRED. J. SLADE.

As already mentioned, the use of a valve-rod link gives us the means of producing considerable variations in the distribution, by altering the position of the fixed centre from the central position which we have now supposed it to have. It will be seen that, if we carry this centre horizontally towards or from the shaft, the arc described by the vibrating end of the suspending link will become inclined, and consequently the path of the block will have the same direction. As this will bring the valve under the influence of the extremity of the link during one stroke of the piston, and under that of a portion nearer to the centre during the other, it follows that the events of the distribution will be retarded in the one case and hastened in the other, thus enabling us to correct the inequalities that exist in the distribution with the central modo of suspension. It is obvious, however, that this mode of rectifying the motion will involve very unequal openings of the valve on the two ends, since, when the link is in its extreme positions, the block will be considerably raised above the normal centre line in one case and depressed in the other. Again, the gain of equality in fore gear will be obtained by a sacrifice of it in back gear, thus aggravating the defects of the distribution there existing. This, it is true, might be obviated by so arranging the centre of suspension of the suspending link that it could be thrown forward or back in reversing, which would certainly involve no great complication. This mode of sustaining the valve-rod link has the disadvantage of increasing the slip of the block; and it is advisable, therefore, not to use it to any very great extent.

The following table shows the distribution produced in full and half gear forward by moving the fixed centre 2 inches toward the crank shaft, and at the same time slightly dropping it, so as to maintain the same central position for the block:—

TABLE SHOWING EFFECT OF MOVING FIXED CENTRE OF VALVE-ROD SUSPENDING LINK 2 INCHES TOWARD CRANK SHAFT.

	Uncorrected Values.				Corrected for Six-Length Rod.				
	Lead.	Greatest Opening.	Suppression.	Exhaust.	Lead.	Greatest Opening.	Suppression.	Exhaust.	
Stroke towards Crank, .	·24	1·12	·73¼	·91¾	·24	1·12	·76½	·93	Full Gear Forward.
Stroke from Crank, . .	·26	1·68	·78	·92	·26	1·68	·75	·90¾	
Stroke towards Crank, .	·42	·64	·44½	·77½	·42	·64	·48¾	·80½	Half Gear Forward.
Stroke from Crank, . .	·42	·80	·51¼	·78	·42	·80	·47	·75	
Stroke towards Crank, .	·25	1·39	·78¼	·92	·25	1·39	·81	·96¾	Full Gear Backward.
Stroke from Crank, . .	·27	1·11	·76	·92½	·27	1·11	·73	·91¼	

As regards the equality of the cut-off on the two strokes, this arrangement produces a very good result in fore gear. The exhaust, however, remains unchanged from the normal motion. This is due to the coincidence of the central positions of the block in the two arrangements. We shall allude again presently to the fact that no variation in the *equality* of the exhaust can be *directly* produced by varying the path of the block when the valve has neither inside lap nor inside lead, though this result may, under certain circumstances, be indirectly produced from such a change by the alteration in the length of the valve rod necessary to maintain equal lead. In back gear the distribution is extremely unequal, and, moreover, the slip of the block in full fore gear is 2¼ inches.

Influence of the use of a Rocking Shaft.—In some engines, to give facility of access to the valve faces, the valve chests are placed on top of the cylinders, instead of at the sides. This is the case with all American engines, and with many outside-cylinder engines on the London and South-Western Railway. In these cases it becomes necessary to employ a rocking shaft to raise the motion from the centre line of the engine to the level of the valve spindle. It will be seen that the motion of the valve will be exactly reversed by the introduction of this part; the eccentrics will therefore require to be set in positions diametrically opposite to those in which they would be placed were the communication with the valve direct, and all the events of the distribution that formerly pertained to the front stroke will now occur on the back stroke, and *vice versâ*. It is obvious that this will make a serious difference in the merits of different arrangements of gear, since those which have given results according with the inequality in the motion of the piston, produced by the action of the connecting rod, will now be in error by double the amount of such inequality, while those whose inequalities were previously increased by the influence of the connecting rod will now give more nearly correct results, and a form of motion well adapted for direct action will therefore be unfit for the use of the rock shaft.

Referring to the tables of results of the different gears, given in the previous chapter, we find that a form previously at variance with the influence of the connecting rod is that in which an open link is employed with the eccentric rods connected behind the centre line of the slot; and this we should accordingly select as a suitable form for use with a rock shaft. Taking the example already given, in which the eccentric rods are connected 3 inches back of the centre line of the slot, and supposing the opposite arms of the rock shaft to be 7 inches long, and the centre of vibration placed perpendicularly above the central position of the block in its vibration from the position at the commencement of one stroke to that at the commencement of the other, we obtain the results given in the following table:—

TABLE SHOWING DISTRIBUTION WITH OPEN LINK ECCENTRIC RODS CONNECTED 3 INCHES BACK OF CENTRE LINE, EMPLOYING A ROCK SHAFT WITH OPPOSITE ARMS 7 INCHES LONG.

	Uncorrected Values.				Corrected for Six-Length Rod.				
	Lead.	Greatest Opening.	Suppression.	Exhaust.	Lead.	Greatest Opening.	Suppression.	Exhaust.	
Stroke towards Crank, . .	·28	1·23	·71	·89½	·28	1·23	·74⅓	·91	Full Gear Forward.
Stroke from Crank, . .	·28	1·31	·76	·91¾	·28	1·31	·73	·90⅓	
Stroke towards Crank, . .	·42	·66	·42⅓	·74⅛	·42	·66	·46⅓	·77⅞	Half Gear Forward.
Stroke from Crank, . .	·42	·69	·60⅓	·79¼	·42	·69	·46¼	·76½	

The distribution effected by this arrangement of gear is thus very nearly perfect, and it is also unattended by the evil of excessive slip of the block produced in some forms of link motion. This general arrangement is the one always used in American engines, the pin on the lower arm of the rock shaft entering immediately into the block, which the use of the open link renders possible. The distribution that would be obtained from any of the other forms of gear for which the results with direct connection have been given, can be found by placing the uncorrected values given for the stroke towards the crank in place of those for the stroke from the crank, and *vice versâ*, and then applying the correction for the effect of the connecting rod as before. If a box link is employed, it will be necessary to employ a connecting link to transmit the motion to the rock shaft, and in this case there will be the slight modification in the distribution due to the mode of suspension of this link, as in the case with direct action which we have already considered. With the open link, also, the same modification will be produced by the arc in which the lower end of the rock shaft moves, as in the case of a valve-rod link suspended from above.

An effect similar to that produced by moving the centre of suspension of the valve rod horizontally may be produced in the case of the rock shaft by inclining the arms towards each other, instead of placing them exactly opposite one another. The effect, however, produced in this case will be less important in respect to the cut-off, and more noticeable in the time of the exhaust, than in the case of the direct connection. The reason of this will be understood from Fig. 170. In this the lines 7 and 1 represent a portion of the centre line of the link at the commencement of the opposite strokes, a' and b' the positions of the same at the time of cutting off with a rock shaft of the usual form, and e' and e'' the positions in the same case at the moment of opening of the exhaust. The extremities, 1 and 7, of the curved path of the upper arm of rock shaft show the positions of the valve stem at the commencement of the strokes. Now, in order that the lead may be equal on the two ends, the valve stem must be made of such a length as to bring the central position of the valve exactly midway between these points, or at e; and it will be seen that, owing to the inclination of the arc described by the lower arm, this central point e will not now fall at the intersection of the lines e' and e'', but at e, a short distance nearer to No. 1; the exhaust, therefore, will take place earlier on that stroke, and later on the opposite one. This, then, is one of the cases alluded to above, in which a change in the path of the valve connection *indirectly* brings about a change in the equality of the release from the necessity for varying the length of the valve rod. That this cannot be done directly by any variation of the path of the block will be apparent from the figure, since, as the exhaust takes place at the same point of intersection on both strokes, inclining the path about this point as a centre, will produce no effect, while raising or lowering it will retard or hasten it equally, or sensibly so, in both strokes, since the angles of inclination of the link on the two strokes are sensibly equal.

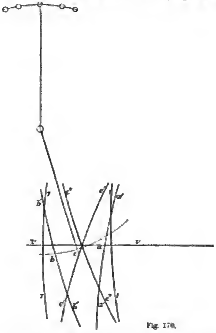

Fig. 170.

Again, as already stated, inclining the arms of the rock shaft does not greatly affect the cut-off. The points a and b on the upper curve, separated by twice the lap of the valve, will be those at which the cut-off will occur; and, laying these off on the lower arc, we see that the inclination of that portion of the arc near a prevents this point falling far from the line a', while the horizontal direction of the part near b also keeps that point from falling far behind the line b'. In the case of the direct connection, these distances, instead of being measured on this curve, would be measured horizontally, and therefore a much greater influence would be exerted in the cut-off in the case of the position *a*. The results for a motion in which the arms of the rock shaft are each inclined backward 1 inch are given in the following table:—

TABLE SHOWING DISTRIBUTION WITH ROCK SHAFT IN WHICH BOTH ARMS ARE INCLINED BACKWARD 1 INCH, LENGTH OF ARMS 7 INCHES, UPPER ARM VERTICAL AT MID THROW.

	Uncorrected Values.				Corrected for Six-Length Rod.				
	Lead.	Greatest Opening.	Suppression.	Exhaust.	Lead.	Greatest Opening.	Suppression.	Exhaust.	
Stroke towards Crank, . .	·26	1·19	·71	·90¼	·26	1·19	·74½	·91¾	Full Gear Forward.
Stroke from Crank, . .	·26	1·50	·76½	·91	·26	1·50	·73½	·89¾	
Stroke towards Crank, . .	·46	·62	·41	·75	·46	·62	·45	·78¼	Half Gear Forward.
Stroke from Crank, . .	·46	·78	·52	·78¼	·46	·78	·47½	·75¼	

It will be seen that the lead is somewhat increased in this case over that which is obtained in the other forms: this is due to the greater length of the path of the block due to its inclination, the full motion due to this arc when placed horizontally being communicated to the valve by the upper and vertical arm of the rock shaft in this case, while in the others that have been considered it was only the horizontal component of the inclined arc that was imparted to the valve. From the same cause also the opening of the valve on the stroke from the crank is greatly increased.

Influence of Mode of Connecting Eccentric Rods.—Hitherto we have in all cases supposed the fore gear eccentric rod to be connected to the top of the link, and the back gear rod to the lower centre. It is, of course, possible, however, to reverse this mode of connection, and so make the lower end of the link the one for fore gear working, and the upper for back gear. The most notable effect produced by this change will be that the lead, instead of increasing toward mid gear, will decrease as we approach that point, and may indeed wholly disappear. The reason for this, as well as the amount of the increase with the one arrangement of rods, or the decrease with the other, will be apparent from a simple geometrical calculation. In Fig. 171. let *d* and *e* represent the positions of the centres of the eccentrics when the crank is on the back centre, and *d'* and *e'* their positions when on the forward centre. Let the length of the eccentric rods be represented by L, and the length of the link from one eccentric rod joint to the other (supposing the case of a box link) by 2 *c*. The linear advance of the eccentrics will be measured by *a*, and the distance of their centres from the centre line by *b*. Suppose, in the first place, that the eccentric rods are connected in the manner which we have so far adopted, which is the usual one, then, when in full gear, the point *f*, which represents the centre of the eccentric rod joint, will be removed from the centre of the shaft by a distance equal to the advance of the eccentric + the length of the eccentric rod — the versed sine of the angle by which the eccentric rod

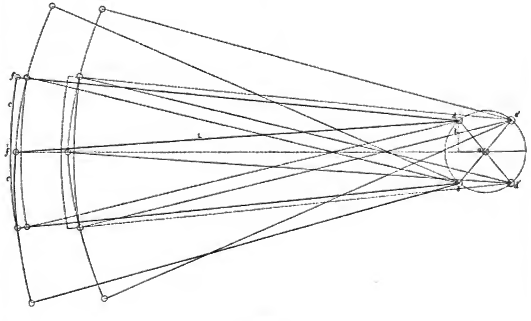

Fig. 171.

is inclined to the centre line. For small angles, such as those which occur in the link motion, this latter quantity will be accurately represented by $\frac{b^2}{2\,L}$.[*] If now we suppose the link to be placed in mid gear, the end of the eccentric rod takes the position *f*, and its distance horizontally from the centre of the shaft becomes equal to the length of the eccentric rod + the advance of the eccentric $- \frac{(c-b)^2}{2\,L}$. The centre of the block in the link whose position determines that of the valve is, however, removed by an additional amount due to the curvature of the link; and as this curve is struck with the length of the eccentric rod as a radius, the versed sine of the arc from the end of the eccentric rod to the centre will be represented by $\frac{c^2}{2\,L}$. Employing the letters which we have chosen to represent the various distances, we have for the distance of the block from the centre of the shaft when in mid gear $a + L - \frac{(c-b)^2}{2\,L} + \frac{c^2}{2\,L}$, which by reduction becomes $a + L + \frac{2cb - b^2}{2\,L}$.

For its distance from the shaft in full gear we also have $a + L - \frac{b^2}{2\,L}$; and subtracting this from the distance when in mid gear, we have $\frac{c\,b}{L}$ as the *increase* in the latter case, which of course measures the variation in the lead of the valve.

When the crank is on the opposite centre, and the eccentrics occupy the positions *d'* and *e'*, the distances of the centre of the block in mid and full gear become respectively $L - a - \frac{2\,c\,b + b^2}{2\,L}$, and $L - a - \frac{b^2}{2\,L}$, the difference in this case also being $\frac{c\,b}{L}$, but with the minus sign, since the valve is brought nearer to the shaft by the increase of lead.

If we, in the next place, suppose the rods connected in the reverse manner, as shown by the dotted lines in Fig. 171, we have the distance of the block from the centre of the shaft when in mid gear, with the crank on the back centre,

$$L + a - \frac{(c+b)^2}{2\,L} + \frac{c^2}{2\,L}; \text{ or}$$

$$L + a - \frac{2\,c\,b + b^2}{2\,L}.$$

The distance when in full gear remains unchanged, since the eccentric rod joint falls at *f'*, as in the previous case, and we therefore have, as before, $L + a - \frac{b^2}{2\,L}$ as the expression for this distance. Subtracting this from the expression for mid gear, and we have $- \frac{c\,b}{L}$ as the variation of lead in mid gear, showing, as already stated, a decrease when the rods are connected in this manner. When the crank is on the opposite centre we have again $L - a + \frac{2\,c\,b - b^2}{2\,L}$ and $L - a - \frac{b^2}{2\,L}$ as the distances of the centre of the block from the shaft in mid and full gear respectively, and the difference, as before, $\frac{c\,b}{L}$, but with the position sign, since the valve is farther from the shaft than in full gear, by the amount of the decrease of lead.

We may see, also, from the expressions $L + a - \frac{b^2}{2L}$ and $L - a - \frac{b^2}{2\,L}$, representing the positions of the valve in full gear when the crank is on either centre, that the eccentric rods must be longer by the quantity $\frac{b^2}{2\,L}$ than the fixed distance from the centre of the shaft to the middle point between the positions of the end centres of the eccentric rods when the crank is on centre, and the consequence of this is, that the extreme throw of the valve will be greater when the valve is moved from the shaft than when in the opposite direction, the same being true in the case of a single eccentric valve gear.

In a valve gear having the dimensions which we have assumed, if we suppose the eccentric rods connected so as to cross when the crank is on the back centre, the lead in mid gear will be reduced to ·04 inch. The distribution effected with such an arrangement is shown by the following table:—

TABLE SHOWING THE DISTRIBUTION WITH ECCENTRIC RODS CONNECTED SO AS TO CROSS WHEN THE CRANK IS ON THE BACK CENTRE.

	Uncorrected Values.				Corrected for Six-Length Rod.				
	Lead.	Greatest Opening.	Suppression.	Exhaust.	Lead.	Greatest Opening.	Suppression.	Exhaust.	
Stroke towards Crank, . .	·25	1·30	·76	·92	·25	1·30	·79	·93	Full Gear Forward.
Stroke from Crank, . .	·25	1·42	·76¼	·92	·25	1·42	·73¼	·90¾	
Stroke towards Crank, . .	·09	·44	·48¾	·83¼	·09	·44	·53	·95¾	Half Gear Forward.
Stroke from Crank, . .	·09	·48	·52½	·83	·09	·48	·48	·80¾	

It will be seen that the principal difference in the distribution, aside from the decrease of lead, is the retarding of the exhaust in half gear. The eccentrics being at this time at their greatest distance from the centre line, transferring the ends of the rods from one side of that line to the other, as we do in this case, will materially alter the distance of the link from the centre of the shaft, and this is the basis of this effect in the distribution.

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[*] This may be proved as follows:—If we let *a*, *b*, and *c*, respectively, represent the hypothenuse, base, and perpendicular of a right-angled triangle, then from the well-known properties of that figure $b = \sqrt{a^2 - c^2}$. If *c* is small relatively to the other sides, we may write for $a^2 - c^2$, $\left(a - \frac{c^2}{2\,a}\right)^2$, the difference between the two quantities $\left(\frac{c^2}{2\,a}\right)$ being in such cases insignificant. We then have $b = a - \frac{c^2}{2\,a}$ or $\frac{c^2}{2\,a}$ = versed sine of angle opposite *c*.

Effect of Varying the Length of the Link.—The expression $\frac{b\,c}{L}$ for the increase of lead at mid gear shows us at once that the effect of increasing the distance between the end centres of the eccentric rods will be to increase the increment of lead toward mid gear. This is of course undesirable; but, on the other hand, the longer the link the less will be its angle of inclination when in its most oblique positions, and consequently the more direct will be the strain on the various parts.

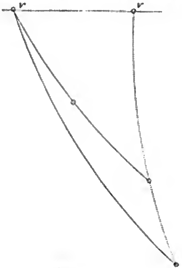

Fig. 172.

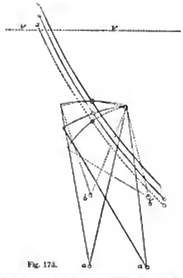

Fig. 173.

Again, by lengthening the link we render it less sensitive to changes in the position of the weigh-bar; and *vice versa*, by shortening it we increase the effect of any change in the position of the shaft. The only effect which the mode of suspension is capable of producing is that of imparting vertical motion to the link. Such motion, of course, produces no effect on the valve when the part of the link in which the block is situated stands vertical and produces a greater and greater influence as the inclination of the link increases. The horizontal distance between the ends of the eccentric rods being nearly a constant quantity for any length of link, it follows that the angle of inclination in any position will be greatest with a short link. This will be apparent from Fig. 172, in which the centre lines of two links of different lengths are shown, both in the position for full gear. The curved dotted line shows the path that the end of the lower eccentric rod would have to describe in changing from one length of link to another. It is evident that any vertical motion of the points of suspension of these links will produce a greater effect on the valve stem V V, in the case of the short link, than in that of the longer one.

The table which follows shows the distribution obtained when using a link of but 6 inches length, with the reversing gear arranged according to the method described p. 166, and also that obtained with the same link when the reversing shaft is moved 2 inches toward the crank shaft, and slightly dropped, the suspending link being attached at the lower eccentric rod joint.

The differences in the distribution produced by this change are thus much greater than those given by a similar movement of the weigh-bar, as shown in the previous chapter.

The effect of any change in the position of the weigh-bar may also be augmented or decreased by shortening or lengthening the suspending link. This will be made clear by an examination of Fig. 173. In this *a* represents the normal position of the lower centre of the suspending link when of the greater length, and *a'* its position when moved a certain distance in a horizontal direction. The arcs described from these centres, and drawn in full lines, show the paths of the upper end of the link in the two cases, and the corresponding positions of the link at the time of cutting off on the stroke from the crank are also shown in full lines. *b* and *b'* represent the positions of the lower centre of a shorter suspending link when in the

TABLE SHOWING THE INCREASED SENSIBILITY OF SHORT LINK TO VARIATION IN POSITION OF THE WEIGH-BAR.

	Uncorrected Values.				Corrected for Six-Length Rod.				Reversing Gear arranged by method of p. 166.
	Lead.	Greatest Opening.	Sup-pression.	Exhaust.	Lead.	Greatest Opening.	Sup-pression.	Exhaust.	
Stroke towards Crank, .	·34	·79	·63¾	·83½	·34	·79	·67¾	·85¾	Half Gear Forward.
Stroke from Crank, . .	·34	1·06	·68¼	·85½	·34	1·06	·64½	·83¼	
Stroke towards Crank, .	·33	·73	·59	·80	·33	·73	·68	·82¾	Weigh-bar moved 2 inches toward Crank.
Stroke from Crank, . .	·35	1·11	·71	·86½	·35	1·11	·67½	·84¼	Half Gear Forward.

normal position, and when removed horizontally by the same distance, as in the former case. The dotted arcs show the motion of the upper end of the suspending link, and the position of the link corresponding with those already given, when the centre of suspension is so displaced, is indicated by the dotted curve. It thus appears that the effect on the valve rod V V, produced by this change in the position of the weigh-shaft, is nearly double, with the short suspending link, that produced with the longer one.

In some cases links have been constructed with the slot curved to a different radius from the length of the eccentric rods. This necessarily involves unequal lead in some positions of the link, but it may aid in correcting the inequalities of the distribution. In effect it is almost precisely the same as varying the length of the eccentric rods in the positions toward mid gear. Thus, if the curve of the link be more convex than that due to the length of the rods, it is evident that the position of the valve in half and mid gear will be the same as if the rods were lengthened by an amount depending on the increase of curvature. The lead will thus be rendered unequal by this amount, and the cut-off will be hastened on the stroke towards the crank, and delayed on the opposite stroke. As the link approaches the position of full gear, these modifications disappear, and the distribution resembles that given by a link of the ordinary curvature. The following table shows the distribution obtained with a link of 3 feet radius:—

TABLE SHOWING DISTRIBUTION WITH LINK OF 3 FEET RADIUS.

	Uncorrected Values.				Corrected for Six-Length Rod.				
	Lead	Greatest Opening.	Suppression.	Exhaust.	Lead	Greatest Opening.	Suppression.	Exhaust.	
Stroke towards Crank, . .	·25	1·25	·76¼	·91¾	·25	1·25	·79¼	·93	Full Gear Forward.
Stroke from Crank, . .	·25	1·31	·75¼	·91¾	·25	1·31	·72¼	·90¼	
Stroke towards Crank, . .	·36	·63	·46¼	·77	·36	·63	·50¾	·80	Half Gear Forward.
Stroke from Crank, . .	·48	·76	·51¾	·80	·48	·76	·47¼	·77¾	

A more uniformly equal distribution would be obtained by simply lengthening the eccentric rods, as in that case the change would affect the distribution in full gear, as well as that in half gear.

CHAPTER X.

THE STATIONARY LINK MOTION.*

This form of valve motion differs from the one just discussed in the manner of changing the connection between the valve rod and the eccentrics. While in the former this is effected by moving the link itself, to bring one or the other of the eccentric rods in conjunction with the valve rod, in this the position of the expansion link is unchanged, but the valve is brought under the influence of either eccentric by moving a jointed link, which connects the valve rod with the block in the expansion link, toward that end of the link to which the eccentric rod, of whose motion it is desired the valve should partake, is attached. This difference in the two motions gives rise to a considerable alteration in the distribution of the steam, as well as to the means of modifying that distribution by variations of the dimensions and positions of the different parts of the motion, and it will now be our object to investigate these changes.

From Fig. 174 the nature of this form of valve gear will be readily understood. It consists of the following parts:—

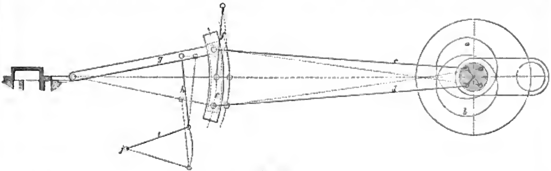

Fig. 174.

1. Two eccentrics: *a*, the fore gear eccentric, *b*, the back gear.
2. The two eccentric rods, *c* and *d* (represented in the Fig. as attached to the centres of the eccentrics), which convey the motion of the eccentrics to the link.
3. The curved expansion link, *e*.
4. The suspending link, *f*.
5. The valve-rod link, *g*.
6. The reversing link, *h*.
7. The reversing arm, *i*, which is keyed on to a weigh-shaft, *j*, worked by levers from the foot-plate of the engine.

The curve of the expansion link is struck with a radius equal to the length of the valve-rod link.

The form of the link may be varied in a number of ways for mechanical convenience, but each change made in the position of the points of attachment of the eccentric rods, or in that of the suspending link, is attended with a variation of greater or less magnitude in the distribution effected by the motion, as we shall presently see. The forms represented on p. 161 are those generally adopted for the stationary link motion.

It will be evident on examining Fig. 174 that the lead will not be affected by any change in the position of the valve-rod link, for as the link stands vertical at the commencement of either stroke of the piston, the block in the link will sweep through the entire length of the slot without moving the valve spindle, to which the other end of the valve-rod link is attached, this point being the centre of curvature of the link in this position. The lead of the valve will therefore be constant at all degrees of expansive working, and this fact constitutes one of the principal points of difference between this form of link motion and the shifting link.

Position of the Eccentrics.—To determine the amount of linear advance to be given to the eccentrics, we must know, in the first place, the following dimensions of the valve motion, viz.:—

The throw of the eccentrics;
The length of the eccentric rods;
The distance between their points of attachment to the link; and
The lap and lead of the valve;

since, unlike in the case of the shifting link motion, each of these elements exercises an important influence on the position in which it is necessary to set the eccentrics to obtain the desired lead on the valve. Moreover, there are two distinct positions in which the eccentrics may be set to produce the same lead, and, as we shall show, nearly the same motion, according as the eccentric rods are connected to the link in such a manner that they shall cross each other (or the centre line of the motion) when the crank is on the forward or when on the back centre. This will be apparent as we proceed to fix the position of the eccentrics, which we shall do as follows:—

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* By Mr. Fred. J. Slade.

Let *a* and *b*, Fig 175, be the positions of the end centre of the upper eccentric rod when the crank is on the forward and back centres respectively. The distance between these points will be equal to twice the sum of the lap and lead of the valve, the valve rod being supposed to be connected in the link at this point. With the length of the eccentric rod as a radius, and these points as centres, describe the arcs *c d* and *e f*, crossing the centre line C C. Draw a line, *g i h*, of a length equal to twice the throw of the eccentric, in such a position that its extremities fall on the arcs *c d* and *e f*, while it is bisected by the centre line C C. The point *i*, in which it cuts the centre line, will represent the centre of the shaft, and *g* and *h* will be the positions of the eccentric when the crank is on the forward and back centre respectively. In this case it will be seen that the eccentric rod will cross the centre line when the crank is on the back centre. If we draw, instead of the line *g i h*, a similar line *j k l*, meeting the arc *e f* above, and *c d* below the centre line, the point *k*, in which it is bisected by the centre line, will represent the centre of the shaft, and *j* and *l* will be the positions of the eccentric when the crank is on the back and forward centres respectively, and the eccentric rod in this case will cross the centre line when the crank is on the forward centre. Drawing the lines *m m′* and *n n′* hrough the points *i* and *k* at right angles to the centre line C C, we have the distance of the points *g* and *h* from the line *m m′* for the linear advance of the eccentric, if the rods are connected so as to cross when the crank is on the back centre, and the distance of *j* and *l* from the line *n n*, for the advance when the rods are connected to the opposite ends of the link. The position of the other eccentric might be found in the same manner, but it is evident that its advance will be the same as that already found for this, since all the conditions are precisely similar, and we need therefore only to lay off the same distance on the opposite side of the centre line for the position of the second eccentric. In the figure the eccentric rods are represented as very short, so as to make the effect of the different modes of connecting them more apparent.

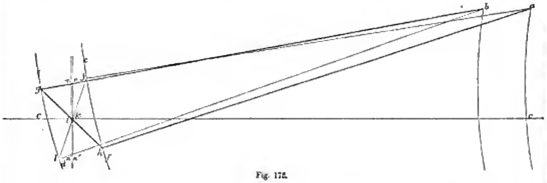

Fig. 175.

It appears from this diagram that when the eccentric rods are connected so as to cross when the crank is on the back centre, the amount of advance necessary to be given to the eccentrics is greater than the sum of the lap and lead of the valve, while, if they are connected in the reverse manner, the advance will be less than this quantity. Although this would at first sight appear to give rise to a considerable difference in the motion of the link, we shall presently show that the positions of the centre line of the link, corresponding with the different positions of the crank, are almost identical in the two cases. The actual distance between the lines *m m′* and *n n′*, or, in other words, the difference in the distances from the shaft at which the link would be situated when the crank was on either centre in the two modes of attachment of the rods, is, in all ordinary cases, but a minute fraction of an inch. Its effect, as far as it goes, is to vary the extreme throw of the valve at the opposite ends of the cylinder.

We will now assume certain definite dimensions for a stationary link motion, and proceed to investigate the motion of the link by diagrams constructed as described on pp. 164, 165, for the shifting link. The dimensions will be:—

Throw of eccentrics, 2½ inches.
Length of eccentric rods, 4 feet 3 inches.
Length of valve-rod link, 30 inches.
Point of attachment of reversing link, 7 inches from centre of block.
Distance from centre to centre of the ends of eccentric rods, 12 inches.
Point to which the suspending link is attached midway between these last-named points, on the centre line of the slot in the link.
Length of suspending link, 15 inches.
Length of reversing link, 21 inches.
Length of reversing arm from centre of weigh-shaft to centre of joint with reversing link, 14 inches.
Lap of valve, 1 inch.
Lead, ¼ inch.
Inside lap, 0.
Width of steam ports, 1½ inches; exhaust port, 3 inches; bridges, 1 inch.

We will also assume for our first purpose that the link is suspended from above, and that the point from which it is suspended is situated vertically over the middle point of the vibration of the suspending link. Also, that the valve-rod link is suspended from below, and in such a manner that the vibration of the reversing link is equal on each side of the vertical when the crank is on each centre and the valve working in full gear forward and back.

The eccentric rods in the first case to be connected to the link, on its centre line, and so that they shall cross when the crank is on the forward centre.

In order that we may determine the path in which the block at the extremity of the valve-rod link will move in positions other than those for full gear forward and back, we must arrange the reversing gear according to the conditions we have assumed. To do this we will let the lines marked 1 and 7, in Fig. 176, represent the positions of the link when the crank is on the back and forward centres respectively, and the lines a, e the positions of the valve-rod link when working in full fore gear, and a', e' its positions when in full back gear, at the same times; the points b and b' show the positions of the extremity of the reversing link where it is attached to the valve-rod link. Describing arcs from the points b, b with the length of the reversing link as a radius, intersecting each other below, we find the position of the lower end of this link d for full gear forward, and in the same manner we find d' for its position in full back gear. Then, with these two points as centres, we have only in the same way again to describe arcs with the length of the reversing lever as a radius, to find by their intersection the position of the weigh-shaft, e. From this centre, then, we can draw the arc $d\,f\,d'$, in which the lower end of the reversing link must always be situated.

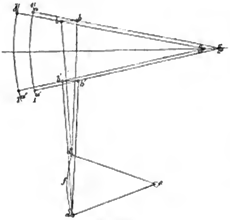

Fig. 176.

Fig. 1, Diagram Plate V., shows the positions of the link during a revolution of the crank. The centre from which the link is suspended is situated at c, and the centre of the link accordingly moves in the arc $a\,b$. The irregular curves described at the top and bottom of the lines representing the positions of the link, mark the path of the end centres of the eccentric rods. The point of the valve-rod link at which the reversing link is attached moves in the arc $d\,e$ when working in full gear forward, and in the arcs shown dotted below this one, when in the various positions of half gear forward, mid gear, and half and full gear backward. This motion of this point causes the extremity of the valve-rod link situated in the link to move in the arc of an ellipse; and this is shown by the line $f\,g$ for the position of full gear forward, and by the similar dotted arcs for the other positions. The weigh-shaft being situated forward of the reversing link, the arc in which the lower end of the latter will move, in changing from full gear forward to full gear backward, will be convex toward the link. When in mid gear, therefore, the lower centre of the reversing link will be nearer the crank shaft by the versed sine of this arc. From the form of the link, the upper centre of the reversing link is also nearer the crank shaft by the versed sine of the arc, described by the point to which it is attached on the valve-rod link in changing to this position, so that the chord of the arc in which the extremity of the valve-rod link moves in the link is maintained nearly horizontal.

The greatest distance by which the line representing the motion of the end of the valve-rod link in full gear rises above the curve marking the positions of the eccentric rods, added to the greatest distance to which it falls below this curve, measures the "slip of the block" in the link; and the amount of this is a point of importance to be considered in a link motion, as it tends to wear a hollow in the surface of the slot of the link at the points where the block is most frequently worked, and is therefore objectionable, although it is sometimes tolerated to a greater or less extent, for the sake of obtaining a more equal distribution of the steam on the two ends of the cylinder. Turning our attention now to the curves representing the successive positions of the centre of the link, we observe, first, that the lines marked 1 and 7, which show the position of the link when the crank is on the back and forward centres respectively, form the boundaries of the figure at the centre line; but as we depart from this point the other lines pass beyond these limits, and form in the aggregate a double concave figure. The distance to which they pass on either side of these first-mentioned lines is, however, not equal, as the distance of the outmost line on the side toward the cylinder from line 1, is noticeably greater than that of the extreme line on the opposite side from line 7. It follows from this, that, with the present path of the end of the valve-rod link at least, the amounts by which the valve will uncover the ports on either end of the cylinder will be unequal, being greatest on the back end.

We also notice that the lines equidistant numerically from the extreme positions 1 and 7, cross each other on the path of the point of suspension. This is due to the fact that, supposing the centre of the link to coincide with the centre line of the motion, the top of the link in any one of two such intersecting positions will have moved the same distance in one direction that the bottom of the link in the other position has moved in the other, and *vice versa*; so that in this case their centres would still coincide, though they would have both moved from their former position. As the arcs described by the ends of the eccentric rods determining the positions of the link in two such cases are always nearly parallel (see Fig. 179), it follows that this intersection will continue to occur sensibly on the line in which the centre of the link moves, even if it be raised or lowered to a considerable distance from the centre line of the motion; and this fact serves as a check upon the correctness of our diagrams.

Motion of the Valve.—Proceeding in the manner described, p. 167, we obtain from the curves representing the positions of the link the motion curves (Fig. 177), showing the position of the valve for each point in the advance of the crank. The following table exhibits the main features of the motion numerically—the points of greatest opening, suppression, exhaust, and compression being given in percentages of the advance of the crank, and the amount of opening of the valve and the slip of the block in the link in inches:—

TABLE I.—STATIONARY LINK MOTION.

	FORWARD.						BACKWARD.			
	FULL GEAR.		HALF GEAR.		MID GEAR.		HALF GEAR.		FULL GEAR.	
	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.
Greatest Opening,	·21	·27	·06½	·12	·00	·00	·07	·11	·21	·25
Amount of Opening,	1·10	1·40½	·50	·61	·25	·25	·49	·60	1·12	1·40
Suppression,	·72¾	·78	·43¼	·54¼	·09	·12	·41	·51¾	·71	·76
Exhaust and Compression,	·91	·92½	·78	·82¼	·16½	·54	·77	·81	·90½	·91½
Slip of Block,	·54		·28		·12		·18		·60	

Before proceeding to examine the character of these results, we will, in the first place, compare them with those obtained from a link motion of precisely the same dimensions, and with the centres of suspension placed in the same relative positions, but with the eccentrics set with the greater amount of advance, the eccentric rods crossing when the crank is on the back centre. The linear advance of the eccentrics in this case will be 1·45 inch, while in the former case it is 1·00 inch.

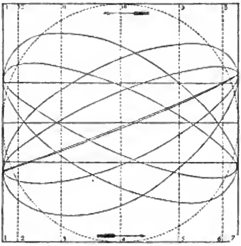

Fig. 177.

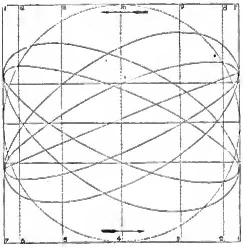

Fig. 178.

Fig. 7, Diagram Plate V., shows the successive positions of the centre line of the link during a revolution of the crank; and Fig. 178 shows in full lines the motion curves of the valve. In order to show more clearly the similarity of this motion to that obtained with the eccentrics set, as in our first example, we have shown in dotted lines the motion curves obtained from the previous arrangement, the fore gear curves in Fig. 177 being, however, placed here in conjunction with the back gear curves, and *vice versa*. The reason of this transposition will be apparent on referring to the diagram of the positions of the link on Diagram Plate V. Supposing for the moment that the centre of the link moved in the direction of the centre line of the motion, instead of in the arc of a circle, it is evident that the upper and lower halves of the group of curves would be exactly alike, since the eccentrics when the link is in position 1 are at equal distances on opposite sides of the centre line; and the motion of one in changing to the position 2 is the same as that of the other in assuming the position 12, but in the opposite direction; and accordingly each position of the top of the link, taken in numerical order, would be removed an equal distance from No. 1, with that of the bottom of the link for the corresponding No. taken in the inverse order. It follows, therefore, that the only difference between the top and bottom halves of the group, in the case of the centre moving in an arc, will be that due to the vertical motion so caused; and by comparing the motion of the valve when moved from the upper end of the link, in the case of the second arrangement of the eccentrics which we are now considering, with the motion obtained from the same end in the first case, we shall include in both cases the effect of the vertical motion of the centre of the link, without any other element of difference in the motion than that due to the effect of the different advance of the eccentrics. The top of the link will, however, in the second case be in connection with the back gear eccentric, while in the first it is connected to the one for fore gear; and hence the propriety of comparing the motion curves in the two cases in this manner.

The groups of curves for the positions of the link are too similar for any difference to be readily detected by the eye; but on laying out the motion curves of the valve, from these we see that a slight difference does exist. Comparing the two sets of curves, we observe that with the second arrangement the valve in fore gear is carried for the positions 10 and 11 to a slightly greater distance from the centre line, and in positions 4 and 5 to a slightly less distance; while in back gear the valve is farther removed from the centre in positions 3 and 4, and nearer in 9 and 10. These are the positions in which the link is most inclined. The inequality in the distance of the valve from the centre in the two cases is partly due to the effect of the inclination of the eccentric rods in bringing the link nearer the crank shaft when the crank is on the centres in the first arrangement of the rods, thereby tending to place the valve farther forward when the working eccentric is near its position of full throw, as pointed out when speaking of the manner of setting the eccentrics. The varying inclination of the rods, however, makes the difference greater at a position not quite at full throw.

Putting the results of this second arrangement into the form of a table, for the sake of numerical comparison with the previous one, we have—

TABLE II.—STATIONARY LINK MOTION.

	FORWARD.						BACKWARD.			
	Full Gear.		Half Gear.		Mid Gear.		Half Gear.		Full Gear.	
	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.
Greatest Opening,	·21	·25	·07	·11	·00	·00	·06½	·12	·22	·26
Amount of Opening,	1·12	1·38	·49	·60	·25	·25	·50	·61	1·10	1·39
Suppression,	·72	·75	·43	·50	·09	·12	·45½	·52¾	·73¼	·77¼
Exhaust and Compression,	·91	·91	·77½	·80	·47	·50½	·79	·82	·91½	·92¾
Slip of Block,	·60		·18		·12		·28		·54	

in which the motion in back gear is to be compared with that in fore gear in the previous table.

From this it appears that the greatest difference in the position of the crank, at the time of the greatest opening of the valve, is but 1 per cent. of the stroke, the greatest difference in the amount of opening but ·02 inch, in the occurrence of suppression 2 per cent. of the stroke, in the exhaust but 1 per cent., and in the compression of course the same, while the slip of the block is identical in the two cases.

The differences tend to render the different epochs more nearly equal in the two strokes with the second arrangement, if we consider only the motion of the crank; but when the effect of the connecting rod is allowed for, we shall find the first the more regular.

If we make the comparison with a link of a different form, we shall get a result which, though varying in a somewhat different manner, will still closely agree with that obtained with the first arrangement of eccentrics. Thus, if we compare the results in the case of a link in which the eccentric rods, instead of being connected on the centre line, have their end centres on an arc 3 inches behind this, and accordingly are made to rise and fall considerably, as the link varies its inclination, we shall find that in the second case the points of closing of the valve and opening of the exhaust are hastened in the stroke towards the crank shaft by about 1¼ per cent., while on the opposite stroke they remain unchanged.

These results indicate sufficiently clearly that, unlike in the case of the shifting link motion, no important difference is produced in the distribution whether the eccentric rods are connected to the link in such a manner as to cross when the crank is on the forward centre, or when in the opposite position, the lead remaining constant in either case, and no other effect of moment being produced. It will therefore be unnecessary to consider this variation further.

As the less amount of advance is the one usually given in practice, the top of the link being used for working in fore gear, we shall adopt that arrangement in our future examples.

As the rock shaft is not in use with the stationary link motion, we shall have no occasion to refer to the distribution produced irrespective of the effect of the connecting rod; we will therefore at once allow for the effect of a connecting rod six times the length of the crank on the motion in Table I., and proceed to examine the merits of the distribution as it actually exists in the cylinder. Table III. on the following page gives the data corrected in this manner:—

TABLE III.—STATIONARY LINK MOTION—MOTION OF LINK. I. CORRECTED FOR SIX-LENGTH CONNECTING ROD.

	FORWARD.						BACKWARD.			
	FULL GEAR.		HALF GEAR.		MID GEAR.		HALF GEAR.		FULL GEAR.	
	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.
Greatest Opening, . . .	·23¾	·23¾	·07½	·10¼	·00	·00	·08	·09¼	·23¾	·21¾
Amount of Opening, . .	1·10	1·40½	·50	·61	·25	·25	·49	·60	1·12	1·40
Suppression, ...	·76	·75¼	·47½	·50	·10¼	·10¼	·45	·47½	·74½	·73
Exhaust and Compression, .	·92¼	·91¼	·81	·79¾	·50¾	·49¼	·80	·78½	·92	·90
Slip of Block, ...	·54		·28		·12		·18		·60	

This, it will be observed, is an example of a practically correct link motion. The greatest opening of the valve is obtained within 3 per cent. of the same point in each stroke: what is of more importance, the suppression occurs always within 2½ per cent. of the stroke at the same point, while the inequality in the exhaust is generally below 1¾ per cent. The greatest slip of the block is but ·6 inch, which is not excessive.

We will now consider by what means we have it in our power to modify these results; and having ascertained the nature of the effect which each change of arrangement of the parts will produce, will proceed to ascertain the exact value of such modifications on the general distribution. It is important, as far as possible, to be able to see *mentally* the effect of each variation in the conditions; and although the motions involved are rather complex, we will point out the influences that are most potent in their effect on the distribution; and a study of the nature of these will assist the understanding as far as possible.

The changes we may make can be divided into two classes—viz., those which affect the positions of the link, and those which alter merely the path of the block. The former are capable of the greater variety, and produce more important results in modifying the motion of the valve. The eccentric rods, of course, determine the position of the ends of the link horizontally, and the only remaining motion that can be imparted is one in a vertical direction. This will affect the position of the valve in two ways: first, by bringing the block nearer to or farther from the position of mid gear; and, second, by drawing the end of the link which is moved away from the centre line nearer to the crank shaft, and throwing the end that approaches the centre in the opposite direction. This latter effect—which is the principal one—will be clearly understood by referring to Fig. 179, which represents the arcs that would be described by the ends of the eccentric rods, in the various positions of the eccentrics, in moving from the centre line to a distance of 7 inches on either side of it. It will be seen that the arcs 1 and 7 have the same inclination on each side of the centre line; and raising or lowering the link through a short distance, when in these positions, will merely alter its inclination without moving its centre horizontally, supposing the rods to be connected on the centre line of the link. The effect of such a change, therefore, would be to alter the position of the valve in full gear without affecting it in mid gear. In the other positions, however, the inclinations of the arcs are unequal on the opposite sides of the centre line. Thus, No. 2 above the centre inclines more backward than No. 1, while below it inclines less; in No. 3 the difference is still greater, and so for all the intermediate positions between 1 and 7. The effect, therefore, of raising the link at any of these would be to carry the top back more than the bottom was carried forward, and the distribution would be affected at mid gear, since the centre of the link would be carried towards the crank shaft. For the positions of the eccentrics between 7 and 1, the arcs are more inclined on the lower side of the centre than above, and, consequently, the effect of raising the link will be to carry its centre forward, while the greatest changes in the position of the valve will be when working in back gear. If we take, for example, the positions 10 and 4, midway between 1 and 7, and suppose the link to be raised in each of these, we see that the upper extremity of the link will be carried back in No. 4 by a much greater amount than the lower end will be carried forward, and, consequently, the closing of the valve will be hastened in forward working on the stroke from the crank in all positions of the block, including mid gear. In No. 10, on the other hand, the upper end will be but slightly carried back, while the centre of the link will be moved forward. As far as this cause is concerned, then, the valve would be delayed in closing in full gear, and hastened in mid gear, on the stroke towards the crank, the amount of the acceleration in the latter case being the same as on the opposite stroke. Now, in the next place, we have to consider the modification of this effect, from the fact that, while the link rises, the block is supposed to remain

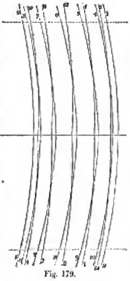

Fig. 179.

stationary. Looking at the position of the centre line of the link in positions 4 and 10 in any of the groups of curves in Diagram Plate V., we see that the approach of the block toward the centre, due to this cause in No. 4, will tend greatly to increase the effect already produced, by carrying the upper end of the link back towards the shaft; while in No. 10 the link is inclined in such a manner as to move vertically in the direction required by the eccentric rods, without compelling the block to move either forward or back. This, then, will tend to increase the difference already existing in the effect of a vertical movement of the link on the distribution at the opposite ends of the cylinder. One more consideration must be noticed in regard to the effect of altering the motion of the link—viz, that as in some such cases the distance between the extreme lines of the group of curves will be increased at the upper and lower ends, it will be necessary to work the block in a position nearer to the centre, in order to maintain the proper maximum travel of the valve, and, consequently, the closing of the valve in such cases will take place earlier in the stroke than would be due to this position of the link, without regard to the change in the position of the block. The reverse will of course be true when the inclination of the link is diminished. If the influence of these three causes, now particularized, can be comprehended, the remaining one—viz., the changing the path of the block across the group of positions of the link—will be readily understood, since a simple inspection of any group will show that, according as the block approaches the centre, the functions of the valve are hastened, and retarded as it moves nearer to the extremities of the link.

The means, then, by which we can affect the distribution of the steam will be, under the first head, viz., by varying the vertical motion of the link and eccentric rods:—

1. Altering the length of the suspending link.
2. Sustaining the link from below, thereby inverting the curve of the path of the centre of the link.
3. By moving the fulcrum from which the link is suspended in a horizontal direction, thereby inclining the path of the centre of the link.
4. By moving the point of attachment of the suspending link forward or back of the centre line, thereby causing the centre of the link to rise when inclined in one direction, and to fall when in the other.
5. By attaching the sustaining link above or below the centre of the link.
6. By connecting the eccentric rods at points not on the centre line—thus causing them to rise and fall as the link varies its angle of inclination.

Under the second head, viz., by changing the path of the block, we have—

7. Suspending the valve rod from above.
8. Moving the weigh-shaft horizontally, thereby inclining the curve described by the end of the reversing link, and therefore that of the block, in the same manner.
9. By changing the length of the reversing link, or by varying the distance from the end of the valve-rod link at which it is attached.

In addition to these means of affecting the distribution, a slight modification may be produced by—

10. Altering the length of the eccentric rods.

There are also other ways in which the distribution may be varied so as to produce a more equal period of admission on the opposite strokes, at the sacrifice of other functions of the valve gear, such as by giving greater lap to one end of the valve than to the other, or by making the eccentric rods of unequal length. As an example of the effect of varying the vertical motion of the link, let us suppose the centre of suspension placed back of the centre line of the link, and trace out the effect upon the distribution. First, then, the upper eccentric rod will be lowered in the extreme position 3, by the inclination of the link, and raised in position 9, thus throwing this end of the link farther from the shaft in the one case, and drawing it nearer in the other, and thereby increasing the distance between the two extremes. The block will therefore require to be worked nearer the centre of the link. Now, in positions 4 and 5, near which the closing of the valve in the stroke from the shaft must occur, the link is still inclined forward, and consequently lowered, and thereby removed farther from the shaft; the closing of the valve will therefore be delayed; nor does the fact that the block is worked nearer the position of mid gear neutralize this effect, since, at the positions in which the valve closes, the link is even more inclined and lowered than in the extreme positions 3 and 9, and consequently the block will stand at a greater distance from the centre, while at the same time the effect of the lowering of the link, in keeping this end of the link at a distance from the shaft, is at a maximum.

In the positions 9, 10, and 11, near which the valve must close on the stroke towards the crank, the link is also much inclined in the opposite direction, and accordingly raised; but as the angle of the eccentric rod at this time is not great, the end of the link is only slightly moved towards the shaft, while the block coming by the same means still nearer to the centre of the link, the closing of the valve is much hastened. The same remarks apply to the occurrence of the exhaust, as also to the working in back gear. Referring to No. 4 in the following table, No. IV., we see the exact amount of these effects in the case of a link in which the suspending link is attached at a point 3 inches behind the centre line, and we observe that the closing of the valve in full gear forward is hastened by 9 per cent. on the stroke towards the crank, and delayed by 4⅞ per cent. on the opposite stroke; in half gear the acceleration amounts to 16⅞ per cent. on the one end, and the delay to 12⅞ per cent. on the other; while in mid gear the quantities are 3½ per cent. and 11¾ per cent. respectively. In the same manner we might suppose other changes, and follow out their effects on the distribution; but the one just considered will serve as an example of the method by which this may be done; and we will therefore simply give the following table, showing the precise effect of the modifications specified in the various positions in forward and back gear:—

TABLE IV.—STATIONARY LINK MOTION, SHOWING THE DISTRIBUTION OF THE STEAM IN FULL, HALF, AND MID GEAR FORWARD AND BACK, WITH DIFFERENT POSITIONS OF THE CENTRES, AS FOLLOWS:—CORRECTED FOR A SIX-LENGTH CONNECTING ROD.

No.	Conditions.	Functions of Valve.	Forward.						Backward.			
			Full Gear.		Half Gear.		Mid Gear.		Half Gear.		Full Gear.	
			Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.
1.	Link suspended from above and at its centre, valve rod sustained from below as in normal motion, position 7 and Table III.	Greatest Opening,	·23¾	·23¾	·07¼	·10¼	·00	·00	·08	·09¼	·23¾	·21¾
		Amount of Opening,	1·10	1·40½	·50	·61	·25	·25	·49	·60	1·12	1·40
		Suppression,	·76	·75¼	·47½	·50	·10¼	·10¼	·45	·47½	·74½	·73
		Exhaust and Compression,	·92¼	·91¾	·81	·79¾	·50¾	·49¾	·80	·78½	·92	·90
		Slip of Block,	·54		·28		·12		·18		·60	
2.	Link suspended from below.	Greatest Opening,	·26	·22¾	·07½	·11	·00	·00	·07	·10¼	·21½	·24½
		Amount of Opening,	1·12	1·39	·50	·59	·25	·25	·51	·57	1·08	1·46
		Suppression,	·76¼	·74¾	·47¾	·48¾	·10¼	·10	·46	·48½	·71¼	·74¼
		Exhaust and Compression,	·92¼	·91	·81¼	·79½	·49¾	·50¼	·80	·78½	·92	·90¾
		Slip of Block,	·45		·20		·03		·29		·75	
3.	Fulcrum carried 3 inches towards the crank shaft, raising the centre of link ¼ inch above centre line of motion in position 1, and depressing it ¼ inch in position 7.	Lead,	·15	·16	·21	·21	·25	·25	·28	·28	·33	·31
		Greatest Opening,	·26	·24½	·09¾	·12½	·00	·00	·07	·16½	·23¼	·22¼
		Amount of Opening,	1·13	1·37	·50	·54	·25	·25	·48	·66	1·06	1·47
		Suppression,	·77½	·78½	·49¼	·50¾	·11¾	·07	·43¾	·47	·73½	·71¾
		Exhaust and Compression,	·93¼	·92¾	·81¼	·82¼	·51½	·50¼	·80½	·76½	·92	·88¾
		Slip of Block,	·85		·60		·52		·58		·92	
4.	Suspending link attached 3 inches back of centre line of link.	Greatest Opening,	·13¾	·30¾	·04¾	·21	·00	·00	·03½	·19	·14¼	·27½
		Amount of Opening,	·90	1·69	·44	·81	·25	·25	·42	·84	·87½	1·70
		Suppression,	·67	·80	·30¾	·62¾	·06¾	·22	·29	·60¼	·63¼	·77¾
		Exhaust and Compression,	·90½	·92½	·72¼	·84¾	·37	·65¼	·69¾	·84	·89	·92
		Slip of Block,	2·45		2·20		2·16		2·36		2·74	
5.	Link suspended at lower eccentric rod joint. Fulcrum above.	Greatest Opening,	·25	·24½	·09¾	·12¾	·00	·00	·09¼	·06½	·23¾	·20
		Amount of Opening,	1·07	1·44	·50	·60	·25	·25	·50	·56	1·17	1·31
		Suppression,	·75½	·71	·50¼	·49½	·12	·08¾	·47½	·41¼	·76¼	·69¾
		Exhaust and Compression,	·92¼	·90¾	·82¼	·78½	·50¾	·50¾	·80¾	·75	·92½	·89
		Slip of Block,	·50		·62		·54		·43		·37	
6.	Eccentric rods connected to link on an arc 3 inches back of centre line of the latter.	Greatest Opening,	·28	·22	·10½	·07½	·00	·00	·11½	·06¾	·28	·22¾
		Amount of Opening,	1·16	1·31	·53	·54	·26	·25	·53	·52	1·23	1·29
		Suppression,	·78¾	·72¾	·51¾	·44¼	·11¾	·07¼	·53¾	·39¾	·78½	·70¾
		Exhaust and Compression,	·93	·90¼	·84	·77½	·56	·44½	·83½	·74¾	·93	·89¾
		Slip of Block,	·61		·29		·10		·21		·63	
7.	Valve rod suspended from above.	Greatest Opening,	·21½	·24½	·07	·10¼	·00	·00	·07½	·11	·26	·22¾
		Amount of Opening,	1·08	1·46	·51	·57	·25	·25	·50	·59	1·12	1·39
		Suppression,	·71¾	·74¼	·46	·48½	·10¼	·10	·47¾	·48¾	·76¼	·74¾
		Exhaust and Compression,	·92	·90¾	·80	·78½	·49¾	·50¼	·81½	·79½	·92¼	·91
		Slip of Block,	·75		·29		·03		·20		·45	
8.	Reversing lever carried 3 inches towards the crank shaft, causing the block to move in an inclined path.	Greatest Opening,	·25	·22¾	·11¼	·10¼	·00	·00	·07	·11	·21½	·24½
		Amount of Opening,	1·19	1·31	·56	·55	·25	·25	·42	·64	·97	1·56¼
		Suppression,	·76¾	·75½	·50¾	·48¼	·12¾	·08¼	·41	·48¼	·72½	·73¾
		Exhaust and Compression,	·93	·91¼	·81½	·80¼	·52¼	·53	·79¼	·77¼	·91¾	·90
		Slip of Block,	·72		·62		·49		·79		1·48	
9.	Eccentric rods 3 feet in length. Advance of eccentrics 0·92 inch.	Greatest Opening,	·25	·24½	·07	·12	·00	·00	·08	·11	·22¾	·24½
		Amount of Opening,	1·06	1·45	·50	·61	·25	·25	·48	·62	1·08	1·43
		Suppression,	·75	·76¼	·47¼	·50¼	·09¾	·10¼	·44¼	·47¾	·74	·73¼
		Exhaust and Compression,	·92¼	·91¼	·80¾	·80¾	·50¼	·52¾	·79¼	·78¼	·92	·90
		Slip of Block,	·61		·28		·10		·20		·61	

In each case here given, all the conditions, except the one specially considered, are supposed to remain as in the motion No. 1, which we have assumed as our normal one, and hence we have the means of observing the effect of each change by comparison with the results of that motion.

It will be seen that the arrangements giving the most equal distribution of the steam on both ends of the cylinder, in all gears forward and back, are Nos. 2 and 7, in which the fixed centre from which the link is suspended is situated on the same side of the centre line of the motion as the reversing lever, and the suspending link is attached at the centre of the link. These two cases are the same in all respects, with the simple difference that the distribution in back gear of one is that for fore gear of the other, since we may consider one motion as but the other inverted. Fig. 180 shows the motion curves of the valve for this arrangement. The most unequal distribution by far occurs with the link suspended as in No. 4, 3 inches back of the centre line, and this indicates at once the powerful effect that a change in the position of the suspended centre horizontally will produce upon the functions of the valve. The slip of the block is also extremely large in this case, amounting to 2¼ inches. As we have already seen, when examining the effect of this change by a study of the causes which determine the peculiar variations produced by modifications of the arrangement, this one is most potent in shortening the period of admission on the stroke towards the crank, and lengthening it on the opposite stroke. In full gear forward the difference amounts to 13 per cent. on the two ends; in half gear it is as much as 32 per cent., or the duration of admission is more than twice as great on one end as on the other; while in mid gear it is 14¼ per cent., the valve remaining open more than three times as long on the stroke from the crank as on that towards it. In back gear the differences are very nearly the same. The motion curves for this case, Fig. 181 exhibit in a marked manner these peculiarities. In No. 6 we have the distribution affected in exactly the reverse manner, by connecting the eccentric rods at points 3 inches behind the centre line of the link, while the suspended centre remains on that line. Fig. 182 shows the motion curves for this case. It might at first be thought that this change should produce effects equally great, but in the opposite direction to those in No. 4, since the vertical movement of the ends of the eccentric rods would appear to be as large as in that case. But that it does not, we are quite able to understand from the attention we have already given to the influences at work in producing these effects. In the first place, the actual horizontal distance of the ends of the eccentric rods from the suspended centre of the link is less than in No. 4, on account of the curvature of the link itself, the difference in the two cases being of course twice the versed sine of that portion of the arc included between the ends of the rods. Hence the vertical movement of the latter, for an equal angle of inclination of the link, would be less in the same proportion. Again, in the case of No. 4, the lowering and raising of the link tends to increase the amount of the inclination of the link, and therefore to augment itself; while in No. 6, as the link is raised at the time when in the other case it is lowered, the motion of the link is restricted, and hence the amount of vertical motion is much less. This will be evident on referring again to Fig. 179, showing the paths in which the ends of the eccentric rods must move when lifted or depressed.

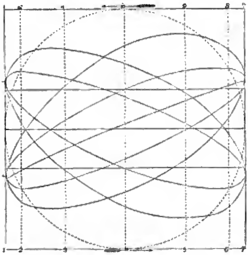
Fig. 180.

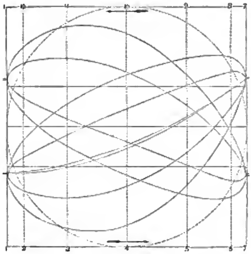
Fig. 181.

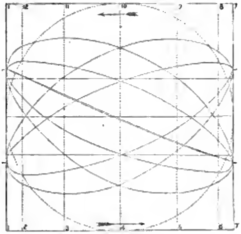
Fig. 182.

But, besides this, the large inequality produced by moving the suspended centre backwards was further increased from the fact that by the vertical motion of the link the block was brought nearer the position of mid gear in the stroke towards the crank, and farther from that position on the opposite stroke. In No. 6, however, there is no vertical movement of the centre of the link, and hence only the first of these causes (and that, as we have seen, in a lessened degree) is operative. We should find, therefore, that if a link were constructed with the three centres all in one arc, parallel to the centre line, the effect of the changed position of the suspended centre would still be strongly manifest, since it would continue to give rise to a vertical motion of the slot of the link. Were the centres placed in advance of the centre of the link the same would be true, but the effect would be reversed, the admission being for a longer period on the stroke towards the crank. The motion curves, Fig. 182, show that in this case the functions of the valve, taken with reference to the advance of the crank merely, without any correction for the length of the connecting rod, are almost exactly equal throughout for both strokes, the amounts of opening, as well as the time at which the various functions occur, being more uniform than with any other arrangement which we have considered.

Referring again to Table IV., we find that a very equal distribution may be obtained by moving the fulcrum from which the link is suspended a certain distance towards the crank shaft, as in No. 3, in consequence of which the centre of the link moves in a path inclined to the centre line. This motion is particularly good in fore gear, as the suppression and exhaust are not only very nearly equal for the two strokes, but the time and amount of greatest opening are also more nearly so than with the former position of the fulcrum. In back gear the amount of opening is more unequal, but the full throw of the valve is still reached at almost the same point of each stroke when in full gear. This change alone, of all that we have considered, produces a variable amount of lead on the valve. Thus we find it to be ·17 of an inch in full gear forward, increasing towards mid gear much as in the shifting link motion, but continuing to increase in back gear, being at a maximum in full gear backwards when it amounts to ·31 inch. This would certainly be an objection to such an arrangement where engines were required to work in both directions; but it might in some cases be adopted with advantage. The decrease of lead towards full gear forward must, we think, be considered rather as a good than a bad feature, since engines are rarely worked at a high speed in full throw of the valve, and an excess of lead is often an annoyance in starting a heavy train. Fig. 183 represents the motion of the valve in this case.

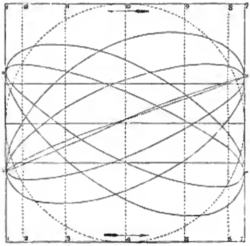

Fig. 183.

No. 5, in which the centre of suspension is placed at the lower eccentric rod joint, gives us the means of observing the effect produced by moving this point in a vertical direction. That the effect of this change would be slight in comparison with a horizontal displacement of the same centre, we should naturally expect, when we consider that it gives rise to no vertical movement either of the centre line of the link or of the ends of the eccentric rods beyond that which arises from the longer arc in which the suspending link vibrates. Accordingly we find that the greatest alteration produced is 5¼ per cent. in the time of closing of the valve in half gear backward, while in other respects the differences are even less. The tendency of this change is to increase the period of admission on the stroke towards the crank shaft, and to diminish it on the opposite stroke. The motion curves are shown in Fig. 184.

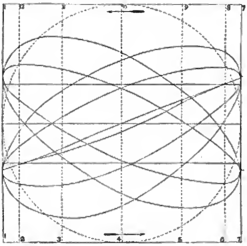

Fig. 184.

From No. 7 we observe that the effect of suspending the valve-rod link from above is to shorten the period of admission on both strokes in full gear, and to lengthen it in back gear, affecting the exhaust also in a similar manner. The cause of this is apparent on a simple inspection of the group of curves, since the path of the block will now be convex toward the centre, or position of mid gear, when in fore gear, thus bringing the block nearer the centre when towards the middle of its motion, at which points the cut off and exhaust occur. In back gear, however, it will be brought nearer the position of full gear at those periods, from the same cause.

No. 8 shows a method by which the amount of opening of the valve may be equalized on the two ends of the cylinder for working in fore gear, at the expense of the uniformity in back gear. It varies the amount of the opening without much corresponding effect on the time of suppression in full gear, but it also increases considerably the slip of the block, which it is desirable always to avoid. Fig. 185 shows the motion of the valve in the various positions of the reversing lever.

Again, in No. 9 we have the effect of varying the length of the eccentric rods. These are supposed in this case to be reduced to 3 feet in length, which is as short as they could well be made; yet we find the difference in the distribution only amounts at its maximum to 2¾ per cent. in the time of reaching the greatest opening of the valve, the amount of opening being varied but ·04½; to but 1 per cent. in the time of suppression in full gear; and to as little as ¼ per cent. in half gear; while the effect on the exhaust is even less. The tendency of shortening the rods is to hasten the functions on the stroke towards the shaft, and to delay them on the opposite stroke. The slip of the block is not much altered. The motion curves are given by Fig. 186.

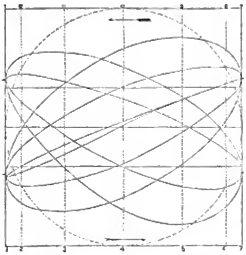

Fig. 185.

In addition to these modifications of the arrangement, it is obvious that there are many others which may be devised that will to a greater or less extent vary the distribution in some or all of the positions of the block. Thus, if we place the weigh-shaft towards the crank, instead of forward, the arc described by the extremity of the reversing lever will have the direction of its convexity reversed, and accordingly the path of the block will be inclined in the positions towards mid gear.

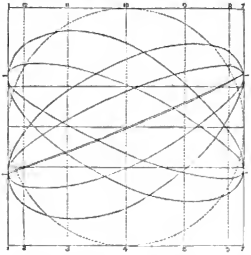

Fig. 186.

The changes that we have considered will, however, be sufficient for our purpose, which is rather to indicate the means by which the valve motion may be modified, and to bring to light the causes on which these means depend for their effects, than to trace out the effect of every conceivable form and arrangement of link motion.

Comparison with simple Eccentric Valve Gear.—It may further assist us in comprehending the influences at work in a link motion if we compare the results obtained in the cases we have supposed with those which should be given by a single eccentric. For this purpose we give in the following table the times at which the valve closes, under the conditions stated in Table IV., in each gear, forward and back, in percentages of the advance of the crank, without regarding the effect of the connecting rod:—

TABLE V.—STATIONARY LINK MOTION—TIME OF SUPPRESSION IN PERCENTAGES OF ADVANCE OF THE CRANK.

	FORWARD.						BACKWARD.			
	Full Gear.		Half Gear.		Mid Gear.		Half Gear.		Full Gear.	
	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.
No. 1, ...	·72¾	·78	·43½	·54¾	·09	·12	·41	·51¾	·71	·76
„ 2, ...	·73	·77¾	·43½	·53	·09	·11¾	·42	·52¾	·70¾	·77¼
„ 3, ...	·74½	·81	·45	·55	·10¼	·08¼	·39¾	·51¼	·70	·75
„ 4, ...	·63	·82½	·27½	·66½	·05¾	·25	·25¾	·64¼	·59¼	·80½
„ 5, ...	·72	·77	·46	·53¾	·10½	·10¼	·43½	·46	·73	·73
„ 6, ...	·75¾	·76¾	·50½	·48½	·10¼	·08½	·49½	·44	·75½	·74
„ 7, ...	·70¾	·77¼	·42	·52¾	·09	·11¾	·43½	·53	·73	·77¾
„ 8, ...	·73½	·78¼	·46½	·52¾	·11	·10	·37	·52½	·69	·76¾
„ 9, ...	·71½	·79	·43¼	·54½	·08½	·12½	·40¼	·52	·70½	·76¼

It will be seen here that in nearly every case the period of admission continues for a longer time on the stroke from the crank than on that towards it, whereas we know that with a simple eccentric valve gear the distribution is exactly equal for both strokes. This affords us another illustration of the effect of the mode of suspension of the link. Did the end of the eccentric rod move always in a straight line parallel to the centre line of the motion, and were the block always to be found in that same line, it is evident that the motion in full gear with a link could not differ in any respect from that with a single eccentric. But in these cases neither of these conditions holds true. The link being suspended at a point in the centre of its curved slot, it will be seen that the end centres of the eccentric rods are removed horizontally from that point by an amount equal to the versed sine of the arc of the link included between them. It results from this, therefore, that the ends of the rods will be unequally lowered when the link is inclined forward and back—that is, the eccentric rods approach nearer to the centre line of the motion when the link inclines from the shaft, than when it stands in the opposite direction. This, as we already know from Fig. 179, will tend to draw the end of the link so inclined farther from the shaft, and therefore to retard the closing of the valve in that stroke. The same increased inclination brings the block nearer to the end of the link, and so augments this effect. When the link is inclined in the other direction, the end of the eccentric rod stands farther from the centre line, and therefore nearer to the shaft, but by a less amount than in the last case, on account of the different angles of arcs 5 and 11 (see Fig. 179); and this would also tend to retard the closing of the valve on this end, were it not that the same cause brings the block nearer the centre of the link, and therefore neutralizes the effect. Now, if we look at No. 5 in full gear back, we find that the distribution is exactly equal. But in this case the suspending link was attached at the lower eccentric joint, and therefore the motion of this point between its corresponding positions at each end of its vibration was parallel to the centre line of the motion. As a consequence, therefore, we have the effect of a simple eccentric gear.

In No. 6, again, we see that the effect of the vertical motion of the overhung eccentric rods is just sufficient to counteract the effect of the unequal rise and fall of the slot of the link in its opposite inclinations. Thus, Fig. 179 shows us that the top of the link in position 5 would be very considerably moved toward the shaft by raising the eccentric rod. As this rise takes place in that position, the closing of the valve would be materially hastened, were it not that this part of the slot of the link is thrown down at the same time, owing to the distance of the suspended centre back of the line connecting the ends of the slot, and therefore the effect of the rising of the eccentric rods is nearly neutralized. On the opposite stroke we see from Fig. 179, again, that the lowering of the rods that occurs in position 11 will not greatly affect the position of the link; but as the upper part of the slot is about at its highest point, the block comes a little nearer the centre, and the closing of the valve is accordingly hastened by about the same amount as on the opposite stroke, maintaining therefore an equal distribution.

Were we to place the suspended centre of the link on a direct line between the ends of the eccentric rods, we should obtain a nearer resemblance to the motion of a single eccentric at full gear; but as we approached the position of mid gear the slip of the block would increase, though the rise and fall of the eccentric rods would necessarily remain equal during each half of the vibration.

We find, however, that the inequality arising from placing the suspended centre on the centre line of the slot in the stationary link is exactly suited to counteract the effect of the connecting rod, and hence this is the position most usual in practice. In the case of the shifting link, the curvature of the link being in the opposite direction, we obtain different results, as has been already shown.

The great variety in the nature of the modifications which we have found to attend on varying the position of the various centres shows us that it is quite possible to combine several of these changes in such a way as to produce a perfectly uniform distribution of the steam. There is a certain position, also, for each centre, in which its effect on the motion is most advantageous; and this may be discovered approximately by an examination of the results given in Table IV.; and a single trial with the centre placed near the point thus indicated will serve to determine the exact position which it should occupy. Thus with regard to the *influence of the mode of suspension*,—We find that in Nos. 1 and 2, in which the suspended centre is situated at the centre of the link, the period of admission in half gear, which is the most important, is shorter on the stroke towards the crank than on the opposite one. By moving this centre 3 inches back of the centre line of the link, we find that this inequality is enormously increased, and, accordingly, it is evident that the best distribution would be obtained by moving this centre very slightly forward.

Again, we observe from No. 5 that by moving this centre vertically through a distance of 6 inches, the inequality existing in Nos. 1 and 2 is reversed, the admission being slightly longer on the stroke towards the shaft in fore gear, and considerably longer in back gear. This, then, indicates that a more favourable result would be obtained by placing the centre at some position midway between these.

Carrying the end of the reversing lever towards the crank shaft to the extent specified in No. 8, we find, reverses the inequality in the suppression existing in No. 1 in half gear forward, and aggravates it in back gear. Accordingly, we should find a better position for it somewhat nearer its original position.

We have already noticed how the changes illustrated in Nos. 4 and 6 may be combined so as to produce an equal distribution. The effect of shortening the eccentric rods is to diminish the relative length of the admission on the stroke towards the crank. This may be neutralized by, at the same time, making one of the changes that produce the opposite effect, such as lowering the position of the suspended centre, or, if it be a box link, by moving the joints at which the eccentric rods are attached slightly backward.

We assumed, as a condition for our normal link motion, No. 1, that the chord of the arc described by the upper end of the reversing link in moving between its positions when the crank is on the opposite centres should be horizontal when working in full gear forward and back. It is often, however, preferred to make this chord horizontal in full gear forward and in mid gear, and we will therefore examine the influence of this change.

The effect of this condition will be seen from Fig. 187. We find the positions d and d' of the lower end of the suspending link when in full gear forward and in mid gear in the manner we have already explained; and from these we find the position of the weigh-shaft e. Continuing the arc $d\,d'$ to the position of full gear back, we find that the end centre of the reversing arm will now fall at d'', the position of the same point with the previous arrangement being shown dotted. It is of course always possible to find a centre from which an arc may be described through any three given points, but that centre in this case would be at c', and would require the reversing arm to be 21 inches long, which is longer than would be advisable. If we adopt this arrangement, then, with the weigh-shaft at e, it is evident from the figure that the path of the block will be inclined in full back gear, as shown by the full line $a''\,a''$, the dotted line showing the horizontal path of the block, with the former position of the weigh-shaft as indicated by the dotted centre near e. The effect of thus inclining the path of the block will be to hasten the closing of the valve on the stroke from the crank in full back gear. Referring to No. 1, Table IV., we see that this already occurs earlier on this stroke than on the opposite one in this gear, and, accordingly, the change will cause a still greater inequality. In all other positions of the reversing lever, except that for full gear forward, the lower extremity of the reversing link will be situated farther forward than with the previous arrangement; and, accordingly, the same effect, but in a less degree, will be produced on the distribution in the other positions in back gear, while for fore gear the relative value of the admission on the stroke from the crank will be increased. Referring again to No. 1, we see that in half gear forward this is already the larger; and it therefore appears that, with the present dimensions, our first arrangement will yield the best results. It will be seen, however, that with a shorter valve-rod link, and therefore an expansion link of greater curvature, the position of the lower end of the reversing link in mid gear may be brought, while still fulfilling the condition now imposed, to coincide with its former position, as shown by the dotted centre in Fig. 187. In the present instance the valve-rod link would require to be about 19 inches long to produce this result.

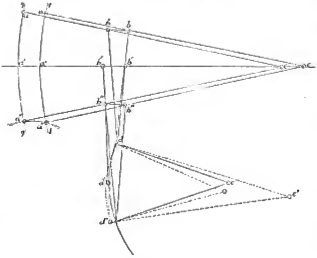

Fig. 187.

We might also have assumed that the chord of the arc described by the upper extremity of the reversing link, in its full motion on either side of the vertical, should be horizontal, instead of having this direction only for that portion of its vibration corresponding to the position of the crank on the centres, and this, again, would modify the position of the weigh-shaft and the distribution in half and mid gear. The motion of the link being in general greatest on the stroke from the crank, the curved path of the block falls lower on that side of the group of curves for the positions of the link, and to bring this horizontal would therefore throw the lower end of the reversing link forward for its positions in full gear; and this might even bring the inclination of the path of the block in mid gear in the opposite direction, the weigh-shaft, of course, moving forward in the same manner as the end of the reversing link. In the case of link No. 1, it would bring it just horizontal.

We have endeavoured, throughout this discussion of the stationary link motion, to place the several primary causes which determine the motion of the valve prominently before the mind, in order that the nature of the motion might be clearly comprehended and retained in the memory. We are aware that the complexity of movements is so great as to render this object difficult of attainment; but there are many to whom the barren statement of a table of results, or the presentation of a complicated mathematical formula, gives but an unsatisfactory solution of the nature of the system; and these will not consider the mental labour necessary to fix in the mind the effective causes in operation as useless, provided they may, by means of it, arrive at a clearer comprehension of the principles underlying the effects of all changes which may be made in the link motion.

CHAPTER XI.

THE STRAIGHT LINK MOTION.*

We have seen that in the case of the shifting link the slot in which the block at the extremity of the valve rod moves is formed in a curve, whose concavity is turned toward the shaft, while in the stationary link the curvature is in the opposite direction, the centres around which motion takes place in the two cases being situated on opposite sides of the link. It would therefore appear that if the two motions were made to take place simultaneously in certain proportions, dependent on the distance of the centres from the link, the slot would require to be curved in neither direction—thus simplifying very materially the construction of the link. The feasibility of so constructing a link motion has been shown by Alex. Allan, Esq., who has obtained a patent for the arrangement. At a later date, but probably without knowledge of Allan's invention, Trick, in Germany, brought forward the same motion. It is quite probable that from many mechanical advantages which this system possesses it will eventually be more used than either of its predecessors. Fig. 188 shows the general arrangement of the parts. It consists, it will be seen, of

1. Two eccentrics, *a* and *b*, for fore and back gear respectively.
2. Two eccentric rods, *c* and *d*, to transmit the motion of the eccentrics to the upper and lower ends of the link.
3. The expansion link, *e*.
4. The valve-rod link, *f*.
5. The reversing shaft, *g*, on which is keyed the double arm *h i*, and which is moved by a connection from the foot-plate, taking hold of the extremity of the lever *j*.
6. The two suspending links *k* and *l*: the first attached to the valve-rod link at the point *m*, and the other applied to the link at some suitable point.

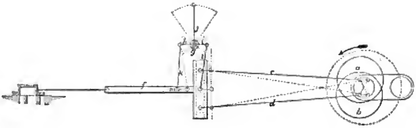

Fig. 188.

It will be observed from the figure that the weight of the valve-rod link *f*, sustained at the extremity of the arm *h*, which, as we shall presently see, is always longer than *i*, tends to counterbalance that of the link and the ends of the eccentric rods acting at the other end of the double lever *h i*. This fact enables the counterbalance weight required in the shifting link motion, which, from the jolting of the engine, brings a certain amount of wear and tear on the parts, to be dispensed with—a point decidedly in favour of this motion.

The angular position of the valve rod being less than in the stationary link motion, the strain on it is more direct, thereby bringing less stress on the reversing gear from the motion of the valve. It is found that the reversing is effected with greater ease than in the shifting link motion, and, as we shall presently show, the slip of the block is somewhat reduced, and consequently less wear occasioned on the surfaces of the slot and block. The link may be constructed in any of the usual forms, according to preference. These points present the principal mechanical advantages of this form of link motion; and it now remains to examine the nature of the distribution of steam effected by it, this being the point of greatest importance in any valve gear.

The first thing we have to determine is *the lengths of the two reversing arms*, *h* and *i*, in Fig. 188, the condition which they are required to fulfil being, that for all relative positions of the link and valve rod the lead shall be found equal on both strokes. In order to arrive at the proper proportion which should exist between them, we will, in the first place, find the positions that the forward end of the valve-rod link will occupy when the engine is on either centre, this point, of course, always corresponding in position with the centre of the valve, since the distance between them is fixed. Having found the middle point between its positions when the crank is on the opposite centres for any one position of the valve rod and link, the problem in hand is to find what proportion must exist between the lengths of the reversing arms, in order that this point may remain the same in all possible elevations of the link. In Fig. 189, let a represent the linear advance of the eccentrics, b the distance of their centres on each side of the centre line of the motion.

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* By Mr. Fred. J. Slade.

L, the length of the eccentric rods;

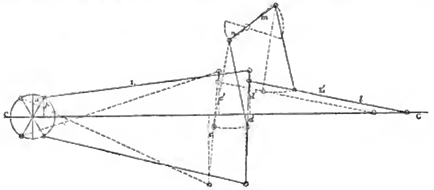

Fig. 189.

L', the length of the valve-rod link;

l, the distance from the forward end of the valve-rod link to the point of attachment of the suspending link;

2 c, the distance between the ends of the eccentric rods;

d, the distance of the centre of the link from the centre line of the motion;

d', the distance of the block at the end of the valve-rod link from the centre line of the motion;

m, the length of the reversing arm connected to the valve rod;

n, the length of the reversing arm connected to the link.

If we suppose the eccentric rods so connected as to cross when the crank is on the forward centre, then, as for small altitudes, the length of the hypothenuse of a right angle triangle exceeds that of the base by the square of the altitude divided by twice the hypothenuse;[*] we shall have for the horizontal distance of the end of the upper eccentric rod from the centre of the shaft when the crank is on the back centre,

$$a + \mathrm{L} - \frac{(c - d - b)^2}{2\,\mathrm{L}}$$

and for that of the lower eccentric rod

$$a + \mathrm{L} - \frac{(c + d - b)^2}{2\,\mathrm{L}}.$$

The difference between these two expressions, $\frac{2\,d\,(c-b)}{\mathrm{L}}$, is the amount by which the link is inclined in any position it may occupy when the crank is on the centre; and the distance, therefore, by which the block will be removed horizontally from the position of the end of the upper eccentric rod will be found from the proportion—

$$2\,c : c - (d + d') :: \frac{2\,d\,(c-b)}{\mathrm{L}} : \frac{\frac{d}{c}\,(c-b)\,(c-d-d')}{\mathrm{L}}$$

Subtracting this from the distance above given for the position of the end centre of the eccentric rod, we obtain for the position of the block

$$a + \mathrm{L} - \frac{(c-d-b)^2 + 2\,\frac{d}{c}(c-b)\,(c-d-d')}{2\,\mathrm{L}} \quad (1.)$$

When the crank is on the forward centre the position of the end of the upper eccentric rod will be

$$\mathrm{L} - a - \frac{(c - d + b)^2}{2\,\mathrm{L}};$$

and for that of the block we shall obtain, in a similar manner,

$$\mathrm{L} - a - \frac{(c-d+b)^2 + 2\,\frac{d}{c}(c+b)\,(c-d-d')}{2\,\mathrm{L}} \,(2.)$$

The arithmetical mean between these expressions for the two positions of the block will give us its central position; and this will be expressed by

$$\mathrm{L} - \frac{(c-d)^2 + b^2 + 2\,(cd - d^2 - dd')}{2\,\mathrm{L}}.$$

To obtain the position of the forward end of the valve-rod link we have to add to this the quantity $\mathrm{L}' - \frac{d'^2}{2\,\mathrm{L}'}$; and we may then write—

$$\mathrm{L} + \mathrm{L}' - \frac{c^2 + b^2}{2\,\mathrm{L}} - \left(\frac{d'^2}{2\,\mathrm{L}'} - \frac{2\,dd' + d^2}{2\,\mathrm{L}}\right) \ldots\ldots (3.)$$

Now, in order that the position of this point may remain constant at all relative positions of the link and valve rod, this expression must be independent of the values of d and d'; and therefore the term in the parenthesis in (3) must be equal to 0.

It is evident from the figure that the following proportion holds good:—

$$d : d' :: n : m\,\frac{\mathrm{L}'}{l};$$

we may therefore write, instead of the quantities containing d and d',

$$\frac{m^2\,\mathrm{L}'}{2\,l^2} - \frac{n\,m\,\mathrm{L}'}{\mathrm{L}\,l} - \frac{n^2}{2\,\mathrm{L}};$$

$$\text{or } \frac{m^2}{n^2} - 2\,\frac{m}{n}\,\frac{l}{\mathrm{L}} = \frac{l^2}{\mathrm{L}\,\mathrm{L}'};$$

whence we readily obtain

$$\frac{m}{n} = \frac{l}{\mathrm{L}}\left(1 + \sqrt{1 + \frac{\mathrm{L}}{\mathrm{L}'}}\right) \ldots\ldots\ldots\ldots\ldots\ldots (4.)$$

In order, therefore, that the lead of the valve may be equal at both ends of the cylinder for all degrees of

[*] See note, page 140.

expansion, it is necessary that the proportion given by this expression shall be maintained between the lengths of the two reversing arms. It will be seen that unless the distance, l, from the forward end of the valve-rod link to the point at which the suspending link is attached is very short, the lever, m, in connection with this point will be the longer. If for any reason this condition be not complied with, we can find from the value of $\left(\frac{d'^2}{2\,L'} - \frac{2\,dd' + d'^2}{2\,L}\right)$ the amount of the inequality in the lead that will be produced by employing any other proportions. Thus, to illustrate this by an extreme case, let us suppose a link motion with the length of the eccentric rods equal to that of the valve-rod link, which we will take as 4 feet; also suppose the proportion between the lengths of the reversing arms to be such that $d = d'$, or the link and block, move through equal distances in opposite directions in reversing, and let the distance between the ends of the eccentric rods be 12 inches. Then in mid gear this expression will be equal to 0, while in full gear, when $d = d' = 3$ inches, it will be equal to $\frac{3}{16}$ inch; and the lead therefore would be greater by this amount at one end than in mid gear, and less at the other end by the same distance—a difference which would be quite inadmissible.

Having now established the relation between the lengths of the two reversing arms, we are prepared to find

The Position of the Eccentrics.—To do this, we must first determine the distance from the centre line at which the ends of the eccentric rods will be situated when working in full gear. If the link is of such a form that the block can coincide in position with the end of the eccentric rod, and we wish it to be so situated when the crank is on either centre, then, of course, the distance of the eccentric rod from the centre line will be equal to d', and $d + d' = c$. The proportion which will exist between d and d' we can easily find, knowing that existing between m and n—it will be $\frac{d'}{d} = \frac{m}{n}\,\frac{L'}{l}$; and having found the value of this for the case in hand, we divide c into two parts, bearing to each other the relation which this bears to unity; the one corresponding with d' will be the required distance. If the reversing shaft is situated on the same side of the centre line as the fore gear end of the link, it will be seen that, if we suppose the block to coincide with the end of the eccentric rod when the crank is on the centre, then when the link stands in its extreme positions the block will be considerably beyond the end of the rod, owing to the combined effect of the inclination of the link and the curvature of the path in which the block is obliged to move, from the manner in which it is suspended. The effect of this will be to increase the throw of the valve beyond that of the eccentrics. As this is generally not desired, it may be necessary to adopt a position for the link nearer to that of mid gear, so as to reduce the extreme throw. The amount of the alteration we can determine as follows:—The vertical movement of the end of the eccentric rod due to its vibration will be nearly enough in all ordinary cases $\frac{(\frac{1}{2}\text{ throw of eccentric})^2}{2\,c}$, since the angular advance is generally not far from 30°. That of the block, in like manner, will be $\frac{(\text{throw of eccentric})^2 - (\text{linear advance})^2}{2\text{ length of suspending link}} \times \frac{L'}{l}$. The sum of these movements must now be divided in the proportion of $n : m\,\frac{L'}{l}$, and the part corresponding with n will be the amount by which the end of the link must be moved from the centre line, in order to maintain the proper travel for the valve. If the reversing shaft be on the opposite side of the centre line, it will be the *difference* of the two motions which will have to be allowed for. Having found the proper distance for the end of the eccentric rod from the centre line, we proceed in a manner altogether similar to that shown in the case of the stationary link. Let a and b, Fig. 17, represent the positions of the end centre of the upper eccentric rod when in full gear forward, at the time that the crank is on the forward and back centres respectively, the distance between them being of course twice the sum of the lap and lead of the valve. From these points as centres, describe, with the length of the eccentric rod as a radius, the arcs $c\,d$ and $c'\,d'$. The centre of the eccentric must of course be found in the first of these arcs when the

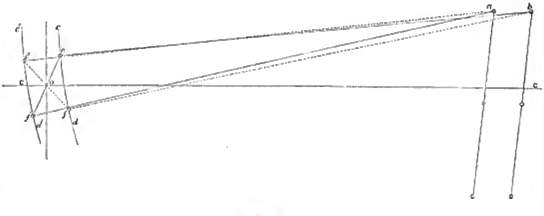

Fig. 100.

crank is on the back centre, and in the other when on the opposite centre. Drawing the line *e f* equal to twice the eccentricity of the eccentric, so that it shall be bisected by the centre line C C, we have the point *o*, at which it crosses the centre line, for the position of the centre of the shaft, while its extremities represent the positions of the centre of the eccentric. Instead of *e f* we may draw *e' f'*, in which case we should obtain the position for the back eccentric when the rods are connected so that they cross when the crank is on the back centre. As in the stationary link, the advance will be greater in the latter case than in the former; but the difference will be less than in that motion, since the ends of the eccentric rod are situated nearer to the centre line C C. Having found the position of one eccentric, the other, of course, is fixed by merely laying off the same distance on the other side of the centre line, on a circle drawn around *o* as a centre, with the radius *o e* equal to the eccentricity of the eccentric.

Our next inquiry is how the lead of the valve is affected in changing from full to mid gear; and a ready answer is furnished by the expressions (1) and (2) which we have already obtained for the positions of the valve when the crank is on either centre. Subtracting the one from the other, we find the distance through which the valve moves to be

$$2a + \frac{2b\,(c-d) + 2\frac{d}{c}\,(c-d-d')\,b}{L}\text{; or}$$

$$2a + \frac{2b(c - \frac{d}{c}(d+d')}{L}. \text{ ....(5.)}$$

If we suppose the valve to be working in full gear when $d + d' = c$, this quantity becomes

$$2a + 2\frac{b\,d'}{L}\text{; ....(6.)}$$

and if we again make d and d' equal to 0, or suppose the valve working in mid gear, we have instead

$$2a + 2\frac{b\,c}{L}, \text{ ....(7.)}$$

which exceeds the expression for the movement in full gear by

$$2\frac{b\,d}{L}\text{; ....(8.)}$$

that is, *With eccentric rods connected so as to cross when the crank is on the forward centre*, the lead will *increase* toward mid gear by half of this amount.

If we repeat the same calculation, supposing the eccentric rods connected in the reverse manner, representing the advance of the eccentrics by a', we should have the quantities

$$2a' - 2\frac{b\,d'}{L} \text{ and } 2a' - 2\frac{b\,c}{L}$$

for the motion of the valve under the same circumstances; and therefore, *when the eccentric rods cross when the crank is on the back centre, the lead is less at mid gear by* $\frac{b\,d}{L}$.

From expression (5) we can find the lead that will result for any intermediate values of d and d' between full and mid gear.

We may observe, also, from these quantities, (6) and (7), the fact which we have already found by our diagram for determining the position of the eccentrics—viz., that the advance of the valve when the rods are connected as in the first case, is greater than that of the eccentrics, and less when they are arranged as in the second case; and of course the position of the eccentrics might thus be determined directly by the formula—

$$a = \text{lap} + \text{lead} - \frac{b\,d'}{L}\text{; or, ...(9.)}$$

$$a' = \text{lap} + \text{lead} + \frac{b\,d'}{L}, \text{ ....(10.)}$$

according to the manner of connecting the rods.

For this purpose we may obtain the value of b nearly enough by assuming, for the moment, the angular advance of the eccentric to be that corresponding to a linear advance, equal to the sum of the lap and lead of the valve as given in the table on p. 157; and the cosine of this angle, taken from a table of natural sines, multiplied by the throw of the eccentric, will give a value of b so nearly correct as to make no appreciable error in the value of $\frac{b\,d'}{L}$.

The increase of lead in approaching mid gear is not so great as in the shifting link motion, since in that it is represented, as we have seen in Chap. IX., by $\frac{b\,c}{L}$ and this quantity, it will be observed, bears the same relation to that obtaining in the present case, viz., $\frac{b\,d}{L}$, as the distances moved through by the link in each case in passing from full to mid gear—a result we should quite naturally expect.

Expressions (4) and (8) also show us that *the increase of lead toward mid gear will be less as the length of the valve-rod link is increased*, since the proportion between the lengths of the reversing levers, m and n, increases with the length of this link, and consequently the distance through which the link is raised, d, is decreased, and with it, of course, the value of $\frac{b\,d}{L}$. If the increased length of the valve-rod link, however, be obtained by reducing the length of the eccentric rods, not much effect will be produced, since a reduced value of L will be more potent in increasing the value of $\frac{b\,d}{L}$ than in increasing that of d by its presence in expression (4), and will therefore tend to neutralize the effect of the diminished value of d due to the increased length of the valve-rod link.

We thus find that the straight link motion combines in its main features the nature of the two of which it is in reality compounded. Before proceeding to examine the distribution of steam effected by this form of valve gear, and the various means by which this may be modified, it is necessary that we should locate the centre of the reversing shaft. From what we have found in the two previous forms, it is evident that the end of each reversing arm should move in a curve, concave in the same direction as that in which the other extremity of its reversing link moves, in order that the vibration of the latter may be equal on each side of the perpendicular in all positions of the reversing lever, and the direction of motion of the valve-rod link, and centre of expansion link, be maintained parallel with the centre line. From the nature of the mode of suspension, however, this is impossible, the reversing shaft being necessarily situated between the points to which the suspending links are attached; and from the short distance between these points, the curves described by the ends of the reversing arms are of much shorter radius than those in which the other ends of the suspending links move, which will further increase the error.

The nearest approximation which we can make to a correct result will be by making the sum of the lengths of the two arms so much greater than the distance between the points at which the suspending links are attached to the moving parts as shall make the error fall in opposite directions when the valve is in full and mid gear. This will make it nearly or quite correct when in half gear both forward and back, and the slight displacement in the other positions will not cause serious inequality in the distribution. Having determined the proportion which must exist between the two arms, according to the formula obtained above, we have only to divide the space between the central positions of the ends of the suspending links attached to the link and valve rod when in mid gear in the same proportion, and repeat this operation for the position of full gear, and from the mean between the points so found (which will never be far apart) erect a perpendicular, and upon this the centre of the reversing shaft will be situated.

We next lay off a parallel line on each side of this, at distances equal to the lengths of the reversing arms, and from the positions of the opposite ends of the suspending links on the centre line when in mid gear, describe arcs, with the length of these links as a radius, cutting these lines. In general there will be a slight space between the two intersections, on account of the sum of the lengths of the arms being greater than the distance between the opposite ends of the suspending links; bisecting this small interval, draw a line from one to the other, and where it cuts the central perpendicular will be the position of the centre of the reversing shaft.

We will now assume certain definite dimensions for a link motion of this type, and proceed to investigate the results to be obtained from it. They will be—

Throw of eccentrics, 2¼ inches.
Length of eccentric rods, 4 feet 3 inches.
Distance from centre to centre of ends of eccentric rods, 12 inches.
Length of valve-rod link, 30 inches.
Distance from forward centre of this rod to centre at which the suspending link is attached, 23 inches.
Length of suspending links, 14 inches.
Sum of the lengths of the two reversing arms, 8 inches.
Reversing shaft placed above the centre line.
Eccentric rods crossed when the crank is on the forward centre.
Lap of the valve, 1 inch. Inside lap, 0.—Lead, ¼ inch.
Steam ports, 1⅛ inches. Exhaust port, 3 inches. Bridges, 1 inch.

In the first place, to ascertain the lengths of the two reversing arms, we find, from the formula (4) giving the proportion between them, that, with the lengths of the rods here assumed, this will be as 1·21 to 1. Dividing 8 inches, which we have taken for the length of the two arms, into two portions, bearing this ratio to each other, we obtain 4⅜ inches and 3⅝ inches nearly enough. The advance of the eccentrics we find, either from the formula (9) already given, or by construction, to be 1·12 inch. The motion curves are shown in Fig. 191, and the positions of the link for all gears in Diagram Plate IV., Figs. 11-15.

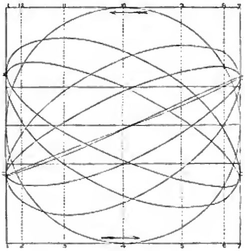

Fig. 191.

Formula (8) shows that the lead will increase ·08 inch in mid gear. The distribution corrected for the effect of a six-length connecting rod is as follows:—

TABLE I.—STRAIGHT LINK MOTION.

	FORWARD.						BACKWARD.			
	Full Gear.		Half Gear.		Mid Gear.		Half Gear.		Full Gear.	
	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.
Lead, ...	·25	·23	·30	·31	·33	·32	·30	·30	·23	·24
Greatest Opening, . . .	·26	·22	·09¼	·09¼	·00	·00	·07	·08½	·29¼	·22
Amount of Opening, . . .	1·14	1·38	·59¼	·61	·33	·32	·58	·67	1·19	1·34
Suppression, ...	·76¼	·72¾	·56½	·46½	·14¼	·11	·51¼	·48¾	·78¾	·74¼
Exhaust, ...	·92¼	·90¼	·86¾	·76¾	·62¼	·47¾	·83	·78½	·93¼	·91
Slip of Block, ...										

These results, it will be seen, are very good; but if we compare them with those yielded by No. 1 stationary link motion, with dimensions as nearly as possible identical, we see that the period of admission on the stroke towards the crank in this case is relatively longer than in that. The cause of this is evidently the fact that in the straight link the suspended centre is directly in line with the end centres of the eccentric rods, while in the other the curvature of the link brings it about ·6 inch back of that position. The powerful influence of any change horizontally in the position of this centre has been fully illustrated in the case of the previous motions, and this suggests at once a means by which a more equal distribution may be obtained in the case before us. The number of variations in the arrangement of the parts which it is possible to make with this form of valve gear is rather more limited than in the previous ones, since the centres of suspension for the link and valve rod, being connected, necessarily move together. It is quite easy, however, to produce a very equal distribution either for fore gear or both forward and back. It is evident that when the weigh-shaft is placed below the centre line the same distribution will be obtained as with the present arrangement, with the exception that the fore gear results become those for back gear, and *vice versa*. We have already seen that, as in the case of the shifting link, if the eccentric rods are connected to the link so as to cross when the crank is on the back centre, the lead will diminish as we approach mid gear; we have found also, in the discussion of the stationary link, that the general effect of this change on the distribution is very slight, and it will therefore be unnecessary to consider this case further here. We give now the results of the more important variations that may be made in the arrangement, assuming in all that the weigh-shaft is situated above the centre line.

TABLE II.—STRAIGHT LINK MOTION—DISTRIBUTION, WITH GEARS ARRANGED AS SPECIFIED, CORRECTED FOR SIX-LENGTH CONNECTING ROD.

			FORWARD.						BACKWARD.			
			FULL GEAR.		HALF GEAR.		MID GEAR.		HALF GEAR.		FULL GEAR.	
No.	DESCRIPTION.	EPOCHS.	Stroke towards Crank Shaft.	Stroke from Crank Shaft.	Stroke towards Crank Shaft.	Stroke from Crank Shaft.	Stroke towards Crank Shaft.	Stroke from Crank Shaft.	Stroke towards Crank Shaft.	Stroke from Crank Shaft.	Stroke towards Crank Shaft.	Stroke from Crank Shaft.
1.	Arrangement as in Table I.	Lead,	·25	·23	·30	·31	·33	·32	·30	·30	·23	·24
		Greatest Opening,	·26	·22	·09¼	·09¼	·00	·00	·07	·08½	·29¼	·22
		Amount of Opening,	1·14	1·38	·59¼	·61	·33	·32	·58	·67	1·19	1·34
		Suppression,	·76½	·72¾	·50¾	·46¼	·14¼	·11	·51¼	·48¾	·78¾	·74¼
		Exhaust,	·92¼	·90¼	·80¾	·76¼	·52¼	·47¾	·83	·78½	·93¼	·91
		Slip of Block,	·85		·27		·22		·27		·39	
2.	Link suspended at lower eccentric rod joint.	Lead,	·26	·24	·30	·31	·31	·33	·30	·31	·24	·26
		Greatest Opening,	·26	·24½	·07	·11	·00	·00	·08	·08¼	·27	·20
		Amount of Opening,	1·15	1·42	·57	·66	·31	·33	·57	·66	1·25	1·28
		Suppression,	·76	·73¼	·50¾	·49¼	·15¼	·12¾	·49¼	·46¼	·79	·72¼
		Exhaust,	·92¼	·90	·80	·77¼	·53¼	·51	·81¼	·77¼	·93¼	·90¼
		Slip of Block,	1·08		·51		·42		·19		·32	
3.	Suspended centre ¼ inch back of centre line of link.	Lead,	·24	·25	·30	·29	·32	·32	·30	·31	·25	·24
		Greatest Opening,	·25	·22	·07	·10½	·00	·00	·07	·09¼	·27	·22¾
		Amount of Opening,	1·12	1·35	·59	·60	·32	·32	·53	·68	1·13	1·34
		Suppression,	·75	·72½	·49½	·47¼	·13¾	·12¾	·48¼	·49¾	·78	·75¼
		Exhaust,	·92	·90¾	·80	·76¾	·51½	·49¾	·81	·79½	·93	·91½
		Slip of Block,	·75		·34		·25		·35		·50	
4.	Centre of reversing shaft carried 1 inch backwards.	Lead,	·23	·21	·29	·30	·32	·32	·31	·30	·25	·26
		Greatest Opening,	·25	·23¼	·05¾	·10¼	·00	·00	·08	·09¼	·27	·20
		Amount of Opening,	1·12	1·36	·57	·60	·32	·32	·62	·68	1·25	1·28
		Suppression,	·75¾	·73	·48	·46¾	·13½	·10½	·53½	·48¾	·79¾	·73¾
		Exhaust,	·92¼	·89¼	·79½	·76¾	·52½	·47¾	·83¼	79	·93½	·90¾
		Slip of Block,	1·00		·32		·16		·27		·67	

We observe from this table that a very good distribution for fore gear is obtained by attaching the suspending link to the lower end of the link at the eccentric rod joint; but the improvement in fore gear is attended with a detrimental effect in back gear. This would probably, on the whole, be a better arrangement than No. 1, in cases where there was little backward running, though, it will be observed, the slip of the block is increased in most of the points of working. A much better motion, however, may be obtained by moving the suspended centre back of the centre line, since its effect is equally beneficial in both fore and back gear. We have given the results for a case in which this centre is moved but ¼ inch from the centre of the link; and with this small change, it will be observed, the difference in the times of closing of the valve on the opposite ends of the cylinder amount to but about 2¼ per cent., the slip of the block, moreover, remaining quite moderate. The motion curves for this case are shown by Fig. 192. A somewhat increased removal of this point would of course render the distribution still more nearly correct.

Again, we may improve the fore gear working by carrying the centre of the reversing shaft towards the crank; but in this case, again, the advantage gained is at the expense of the distribution in back gear. The slip of the block is in general small in this case, and the arrangement quite a feasible one. We observe here the same effect on the lead as in the case of the stationary link, in which the centre of suspension was carried back—viz., a decrease towards full gear forward, and increase towards full gear back, as compared with that obtained when the centre is directly over the centre of the link.

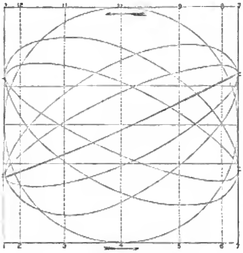

Fig. 192.

An arrangement employing an open link, with the eccentric rods connected behind, would, as we might predict, give a very unequal distribution. Supposing the distance of the end centres of the rods from the centre line to be 3 inches, the times of closing of the valve with a six-length connecting rod would be—

	Stroke towards Crank.	Stroke from Crank.
Full gear forward, . . .	79¼	69¼
Half gear forward, . . .	56	41
Full gear back, ...	81¾	70¾

It would be possible to correct this by at the same time moving the suspended centre back, but the other arrangements are to be preferred. With regard to the causes of these various effects, little need be added here, since they are quite identical with those operating in the case of the stationary link which we have already studied. The cause of the increased inequality in back gear when the suspending link is attached at the bottom will be understood when we observe in the group of curves on Diagram Plate IV., showing the motion of No. 1 link in full back gear, the relative positions of the centre of the eccentric rod and the block at the times of cutting off. In No. 1, on the stroke from the crank, or in position 10, we find the centre of the eccentric rod considerably above its position 1 when opening the valve; and as the effect of attaching the suspending link at this point would be to make this centre remain at all times in the arc described by the suspending link, it would be lowered when in position 10, and the closing of the valve hastened at that time in consequence. Again, the particular effect produced by moving the centre of the reversing shaft towards the crank is traceable to the inclination produced in the path of the block by this means, causing it to rise during the stroke from the crank, and to fall during the opposite stroke. The cause of the great effect produced by moving the suspended centre horizontally has been fully explained in the previous chapter, and it will be unnecessary to repeat those remarks here.

We will conclude the discussion of this form of valve gear by giving an example from practice, in which quite different dimensions were adopted. They were as follows:—Throw of eccentrics, 2½ inches; length of eccentric rods, 4 feet 2 inches; length of valve-rod link, 3 feet 5½ inches; forward centre of ditto, to point of attachment of suspending link, 2 feet 11 inches; centre to centre of eccentric rod ends, 10 inches; length of suspending links, 13½ inches; length of reversing arm for valve-rod link, 4¼ inches; for expansion link, 2⅜ inches; centre of reversing shaft, 4 feet 4$\frac{5}{16}$ inches forward of centre of shaft, or ½ inch forward of the mean position corresponding to the vibration of the suspending link, and placed *below* the centre line of motion; suspended centre on the centre line of link and in the middle of its length; link of box form; lap of valve, 1¼ inches; lead, ¼ inch; steam ports, 1½ inches; exhaust ditto, 3 inches; stroke of piston, 20 inches; length of connecting rod, 6 feet 1½ inches, or a little over seven times that of crank.

The effect of placing the centre of weigh-shaft forward of its normal position, when below the centre line, is of course the same as moving it back when above, since it inclines the path of the block in the same manner. The proportion between the reversing arms is 1 to 1·86, the correct ratio, according to the formula we have obtained (4), being 1 to 1·74. The inequality caused by this slight discrepancy is quite insignificant, as will appear if we find the value of the quantity in parenthesis in expression (3). Formula (8) shows that the increase of lead at mid gear will be ·06 inch; *d*, or the distance of centre of link below centre line when in full gear being 1·56.

TABLE III.—RESULTS OF FULL-SIZED MODEL EXPANSION GEAR, AS ARRANGED FOR GOODS ENGINES.

Notch.	Lead.		Amount of Opening.		Suppression.		Exhaust.		
	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	
8, .	¼	¼	$1\frac{3}{16}$	$1\frac{5}{16}$ +	74	71	92	90	+ Forward Gear.
7, .	¼	¼	$\frac{15}{16}$	$1\frac{1}{16}$	66	64	89	87	
6, .	¼ +	¼ +	¾ +	⅞	58	57	86	84	
5, .	¼ +	¼ +	⅝	¾ —	50	50	83	80	
4, .	¼ +	¼ +	$\frac{9}{16}$	⅝	43	43	80	77	
3, .	¼ +	¼ +	$\frac{7}{16}$ +	½	35	36	76	73	
2, .	$\frac{5}{16}$ —	$\frac{5}{16}$ —	⅜ +	$\frac{7}{16}$	27	28	71	68	
1, .	$\frac{5}{16}$ —	$\frac{5}{16}$ —	$\frac{5}{16}$ +	$\frac{5}{16}$ +	17	17	62	59	
0, .	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	10	10	51	50	
1, .	$\frac{5}{16}$ —	$\frac{5}{16}$ —	$\frac{5}{16}$ +	$\frac{5}{16}$ +	20	17	62	60	– Backward Gear.
2, .	$\frac{5}{16}$ —	$\frac{5}{16}$ —	⅜ +	⅜ +	30	27½	70	68	
3, .	¼ +	¼ +	½	½	41	38½	77	74½	
4, .	¼ +	¼ +	⅝	$\frac{11}{16}$ —	52	49	82	80	
5, .	¼ +	¼ +	$1\frac{1}{8}$ —	⅞	60½	57	85½	83½	
6, .	¼	¼ +	$1\frac{1}{8}$ +	$1\frac{3}{16}$ +	67	64	88	86	
7, .	¼ —	¼	$1\frac{3}{16}$	$1\frac{1}{8}$ +	74	71	92	90	

CHAPTER XII.*

VALVE GEAR OF HEUSINGER VON WALDEGG, OR WALSCHÄERTS' GEAR.

A FORM of valve gear differing materially in mechanical construction from either of the ordinary types of link motion has been used to some extent on the Continent, and is known by the above titles,—Herr Heusinger von Waldegg having introduced it in Germany, and M. Walschäerts in Belgium. It has also been applied to many of the Crampton engines on the Chemin de Fer du Nord, in France. But a single eccentric is employed, and that placed exactly at right angles with the crank. The motion of the valve is obtained from a combination of that of this eccentric and the motion of the piston, by means of an arm partaking of the motion of each. The arrangement will be understood from Fig. 193. The eccentric or equivalent return crank is shown at *a*. The eccentric rod *b* transmits the motion of this to the link *c*, which swings on a fixed centre, *d*, at such a distance above the centre line that the end of the eccentric rod may remain in that line. The link may either be placed forward of the piston cross-head, as shown in the figure, or between that and the crank shaft, the curvature in the latter case being in the opposite direction, so as to remain concave towards the cross-head. A radius rod, *e f g*, equal in length to the radius of curvature of the link works in the slot in the latter, being capable of being raised or lowered by a reversing lever and suspending link attached at *f*. The end *g* of the radius rod is attached to an intermediate lever, *h g i*, the lower end of which is moved by a suitable connection with the cross-head of the piston rod, and the upper end is attached to the valve spindle v. The motion from the cross-head is obtained by carrying down a projection, *j k*, to a distance sufficient to give a suitable length to the lever *h g i*; and this latter may either work in a socket, through which it can slide as the inclination varies, or a connecting link may be provided to allow of the vertical motion of the end of the lever. It will be seen that as the radius rod *e f g* is brought nearer the centre *d*, the motion of the point *g* between its extreme positions will be reduced, disappearing entirely when the block at the end of the radius link coincides with this centre. In the latter case the motion of the valve will be wholly derived from the cross-head. When the radius link is brought into the lower end of the link, it is evident its motion will be exactly the reverse of that which it has when working at the other end; and as the eccentric *a* is set without advance, it is clear that by thus changing the position of the radius link it is made to perform the functions of either a fore or back eccentric. The object sought to be attained in this form of valve gear is to vary the travel of the valve while the lead is maintained constant. We shall clearly understand the manner in which this is effected if we divide the mechanism into two portions—viz., 1. The eccentric and curved link, with their various rods; and, 2. The attachment to the piston by means of the lever *h g i*. It will be seen that the first set of parts would, if acting alone, give the valve a motion exactly the same as would be obtained from an eccentric set without advance, and of any throw from that of the eccentric to 0, according to the position of the block in the link. The connection with the piston is simply to furnish the required advance to the valve when the crank is on the centre, the dis-

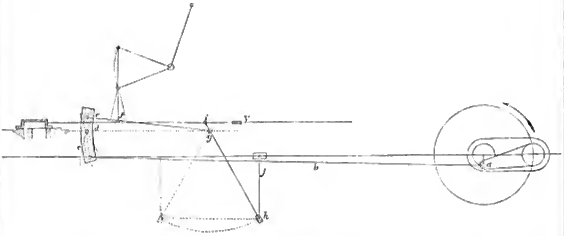

Fig. 193.

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* By Mr. Fred. J. Slade.

tances ig and gh on the intermediate lever being so proportioned as to reduce the length of the stroke to twice the sum of the lap and lead of the valve. The amount of lead depends solely upon the proportions of this lever, and will remain constant under all circumstances, since, as the link is curved with a radius equal to the rod efg, we may place the block in any position in the length of the link without changing the position of the centre g. Although depending for its effect in varying the cut-off on the same cause as the ordinary link motions—viz., a reduction of the travel of the valve, with a constant amount of lap and advance—there is one important difference in the motion produced, arising from the fact that the length of the connecting rod influences the motion of the valve to a greater or less extent, since this is partly dependent on the motion of the piston; while in the previously considered cases the action of the valve depends solely on the movement of the eccentrics and the length of the eccentric rods. As the piston is always nearer to the crank shaft than would be the case were its motion the same as that of the crank, it will be seen from Fig. 193 that this will keep the valve forward of the position which it would otherwise have during both strokes, and thereby retard the closing of the valve on the stroke from the crank, and hasten it on the opposite stroke. This will tend to some extent to compensate for the effect of the connecting rod in producing unequal distribution on the two ends. Furthermore, it will be observed that as we approach mid gear this compensation becomes more complete; and when the rod efg is at the centre of the link the correspondence between the motion of the valve and that of the piston is perfect, since in that case the whole of the motion is derived from the cross-head, and the valve will be found in its central position just at the half stroke of the piston. This fact gives this motion an important advantage over that derived from an eccentric having the same throw and requisite advance, or even to some extent over that derived from a link motion of the usual form, though, as we have seen in the latter case, it is possible, by various modifications of the mode of suspension, or the positions of the different centres, to obtain a motion which shall be adapted to the disturbing influence of the connecting rod. If we employ a connecting link to transmit the motion of the cross-head to the lever igh, it will be seen that it is possible to increase this modifying influence of the connecting rod on the motion of the valve by placing the stud hj back of the lever igh. In such a case the obliquity of the link, supposing it to move only *below* the end of the stud, would in general draw the end of the lever igh still more towards the crank shaft, causing the valve to close too late on the stroke from the crank towards mid gear, and making the distribution more nearly equal towards full gear.

We will first notice the distribution obtained when the lever works in a socket on the cross-head, that we may observe the effect upon the motion of the connecting rod alone.

To construct the motion curves for the valve with this form of gear, we first draw the line mn, Fig. 16, Diagram Plate IV., representing the path of the stud on the cross-head by which the lever igh is moved, and lay off on this six points showing its positions corresponding with those of the crank pin for each 30 degrees of its revolution. Next lay off the positions of the curved link corresponding with the same points in the revolution of the crank, and number them from 1 to 12. In general, since the eccentric rods are very long in comparison with the eccentricity of the eccentrics, each two positions equidistant from the extreme ones 4 and 10 will sensibly coincide. Then draw lines representing the radius link in its extreme and middle positions, and mark on the curves representing the centre lines of the link the position of the centre of the block in each case: a circular curve through these points will represent the path of the block during the entire revolution. Now from each point where this arc intersects the positions of the link, describe short arcs, with the length of the radius link as a radius, in the positions where the back end of this rod must be situated, and number these to correspond with the positions of the link from which they are drawn. With a straight template, representing the lever igh, we may find the points on the line V V, representing the direction of the valve spindle, at which the end of this lever will be situated when passing through each of the divisional points on the line mn, and the arc corresponding in number with it in which its intermediate centre must be situated. These distances, laid off in the usual way, will give us the elliptical motion curve.

From this construction we observe that the extreme positions of the link, Nos. 4 and 10, do not give the extreme positions of the valve; but, owing to the greater inclination of the intermediate lever, these agree in general with Nos. 3 and 9. It follows from this, that either the eccentric must have a shorter throw than is required for the valve, or the radius link must be worked nearer the centre of the link than the point of attachment of the eccentric rod. The exact amount of motion that must be derived from the link will depend on the proportion between the intermediate lever and the length of the stroke, and this, again, is determined by the amount of lap and lead of the valve. Assuming, as is always sensibly the case, that the full throw of the valve occurs in positions 3 and 9, we can easily determine the motion of the link that is necessary, when the proportions of the intermediate link have been decided upon. To do this, we have only to lay off on the line V V a distance equal to the full throw of the valve, and from these points draw diagonal lines through the points 3 and 9 (which are always distant from each other by half the length of the stroke). The distance between a point on each line, representing the centre at which the radius link is attached, will give the motion of the link required between positions 3 and 9, and this will always be ·866 of the full motion, or of the double throw of the eccentric, if this is connected at the point at which the radius link works, since these positions of the eccentric are 60 degrees on each side of the central one.

It is evident from the figure that the dimensions of the parts must be so proportioned that the centre g, when in positions 1 and 7, shall fall upon the line in which the centre of the link is situated, since it is only when so situated that the radius link may be moved to any position in the link without moving the valve spindle, or, in other words, affecting the lead. To show the nature of the distribution effected by this valve gear, we have constructed motion curves for a case in which the dimensions were assumed to be as follows:—

Throw of eccentrics, 2¼ inches.
Length of eccentric rods, 8 feet.
Length of stroke, 2 feet.
Length of connecting rod, 6 feet.
Length of radius link, 30 inches.
Point of attachment of suspending link, 7 inches from centre of block.
Distance from centre of link to end of eccentric rod, 6 inches.
Travel of valve, 4¾ inches.
Lap of valve, 1 inch.
Lead of valve, ¼ inch.
Length of shorter arm of intermediate lever, 2½ inches; and therefore,
Length of longer arm to stud, when at end of stroke, 24 inches. This dimension fixes that of stud on cross-head as 20¾ inches below centre of link, or 14¾ below centre line of engine.
Length of suspending link, 14 inches.

The results corrected for the effect of the connecting rod are as follows, the motion curves being shown by the full lines in Fig. 194:—

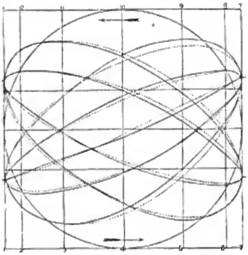

Fig. 194.

	FORWARD.						BACKWARD.			
	Full Gear.		Half Gear.		Mid Gear.		Half Gear.		Full Gear.	
	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.
Greatest Opening, ...	·27	·24½	·07½	·12	·00	·00	·11½	·12	·28	·19
Amount of Opening, . . .	1·34	1·37	·50	·62	·25	·25	·56	·57	1·38	1·31
Suppression, ...	·76¼	·72	·48¼	·48¾	·12	·12	·50¾	·46¾	·78¼	·70¾
Exhaust, ...	·92	·89¾	·79¾	·76¾	·50¼	·49¼	·79¾	·76	·92¼	·89¾

These results are tolerably satisfactory in fore gear, though not perfectly equal for both strokes. It will be remarked, as already noticed, that the distribution accords more closely with the motion of the piston as we approach mid gear. The cause of the greater inequality in back gear will be understood by referring again to the Diagram on Plate IV., where it will be observed that, on account of the more nearly vertical position of the intermediate lever at the time of closing of the valve than when at the end of the stroke, the intersections with the arcs representing the positions of the end of the radius link occur below the centre line of the link, and hence the valve is thrown forward in fore gear and backward in back gear, owing to the inclination of these arcs—thus increasing the correspondence of the motion of the valve with that of the piston in the first case, and diminishing it in the second. It appears from this consideration that a motion more nearly similar in both forward and back gears would be obtained by making the throw of the eccentric larger, and working nearer the centre of the link.

The dotted lines in Fig. 194 show the distribution effected when the intermediate lever is connected to the cross-head by means of a short link. We have taken the length of this as 6 inches, and supposed it placed *back* of the lever, and to be horizontal when at the ends of the stroke—thus throwing the whole of its vibration below that position. The results, expressed numerically, and corrected for the effect of the connecting rod, are as follows:—

	FORWARD.						BACKWARD.			
	Full Gear.		Half Gear.		Mid Gear.		Half Gear.		Full Gear.	
	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.	Stroke towards the Crank Shaft.	Stroke from the Crank Shaft.
Greatest Opening, ...	·22¾	·24½	·05¾	·12	·00	·00	·08¾	·12	·26	·21
Amount of Opening, . . .	1·16	1·39	·50	·59	·25	·25	·52	·59	1·31	1·29
Suppression, ...	·76	·76¼	·41	·50¾	·10¼	·09¾	·46½	·49¾	·79	·74
Exhaust, ...	·92¼	·91¼	·78½	·89	·46	·54	·81	·80	·93½	·90¾

This change, it thus appears, produces too great a reduction of the admission on the stroke towards the crank, except in full back gear. But if the link were placed so as to vibrate both above and below the horizontal, this would not be true to the same extent. The reason of this, and of the effects here produced, will be understood by referring to Fig. 193 and to Fig. 16, Diagram Plate IV. In Fig. 193 we see that the effect of this mode of connection has been to bring the positions of the valve nearer to the centre line towards the ends of the stroke, but that this is much more powerfully effected near the end from the shaft than at the opposite end. Referring to Fig. 16, Diagram Plate IV.—in which, to illustrate the influence of this change, we have shown below the line *m n* dotted arcs representing the path of the end of the link in vibrating from that line—we see that as the ends of the lever would be obliged to coincide with these arcs in the case we are considering, it would be drawn towards the shaft, in all cases from 7 to 11 and 3, but moved towards the centre, though to a less extent, in the remaining positions. If the centre on the cross-head to which the link is attached were lowered, thereby making these arcs less inclined, the tendency to bring the lever nearer the centre would be more equal at the two ends, the effect of which can easily be traced out on Fig. 193.

Supposing this link to vibrate equally on each side of the horizontal, the suppressions would occur as follows:—

	FORWARD.			BACKWARD.	
	Full Gear.	Half Gear.	Mid Gear.	Half Gear.	Full Gear.
Stroke towards Crank,	·78	·46½	·12	·52¾	·81
Stroke from Crank,	·71¾	·45¾	·08	·44½	·71

We see, therefore, that the use of a socket through which the intermediate lever may slide in its vibration gives a much better distribution than is obtained with a connecting link, though the latter, no doubt, is a better mechanical contrivance, as it is subject to much less wear.

In any position in which it may be placed, the use of a link in connection with the end of the intermediate lever will tend to hasten the closing of the valve, when that occurs before half stroke, and to retard it when it occurs after that period, and to diminish its maximum travel by lessening the divergence of the positions of the lever at Nos. 9 and 3. The greater the length of the lever, however, the more nearly will the effects of the two modes of connection resemble each other.

The motion may be modified by removing the fixed centre of the link back of the centre line of the slot. This will increase the regularity in back gear, by retarding the closing of the valve on the stroke from the shaft, and hastening it on the other stroke, and will not very materially affect the distribution in fore gear; its effect differing somewhat in this respect from that produced by the same change in the case of the ordinary link motions. In these, both ends of the link being governed by eccentric rods considerably inclined, the vertical motion produced by placing the suspended centre back of the centre line increases materially the inclination of the link. In this motion, however, the eccentric rod is not much inclined, and a vertical movement of the centre causes the link to assume a position nearly parallel to its former one, but farther removed from it at the back gear end, on account of the necessarily accompanying movement of the centre of the slot horizontally, tending to neutralize the displacement as we approach the top of the link. The principle on which this valve gear is constructed is highly ingenious, and the distribution of the steam that may be obtained from it quite good; but the obvious objection to its use is a lack of simplicity in the arrangement, and a certain amount of difficulty in its application to the usual types of engines. It is usual to employ a return crank for the eccentric motion with outside cylinder engines, to which it is generally applied; but the same motion could of course be obtained from an eccentric on the shaft, by employing a rock shaft to transmit it to the link. It would be possible, also, to substitute an eccentric having the same motion as the crank in place of the stud on the cross-head, in which case, of course, the intermediate lever would be made short; but unless the arrangement were placed quite close to the shaft, and a long valve spindle employed, the compensating effect of the connecting rod would be lost, and the distribution vitiated. None of these arrangements would be desirable; nor does the gear compare in simplicity with the link motion as usually constructed, which yields, moreover, quite as good a distribution. We have considered it chiefly on account of its wide difference from the other gears in the means by which constant lead is combined with varying travel, and also because its use on the Continent, where outside valve gears are not objected to, but in many cases even preferred, entitles it to a place in a work of this kind.

CHAPTER XII.*

REVIEW OF THE PRINCIPLES OF THE SLIDE VALVE.

THE slide valve, as applied to the locomotive, combines in one simple mechanism the means of admitting, suppressing, and releasing the steam to and from the cylinder.

If constructed without lap or lead, the steam will be admitted from the moment of the commencement of the stroke to the very end, and will begin to escape only when the piston commences its return stroke.

This being wasteful in the consumption of steam, and detrimental to the power exerted by the engine, from the difficulty with which the steam effects its escape after having done its work, means are provided for suppressing the admission of steam before the piston has completed its stroke, and of releasing the steam, also, before it begins to return.

The addition of lap to the valve, at the same time setting the eccentric so that the valve shall open at or before the commencement of the stroke, accomplishes both these objects.

It is usual, also, to give a certain amount of lead to the valve, in order that the steam may be promptly admitted and the port opening be wide enough, as the piston advances, to allow the pressure to be well sustained. The amount of this varies with the speed of piston at which it is intended the engine shall work, and also, to a certain extent, with the opinions of engineers, ¼ inch being a usual amount in this country, and ⅛ inch, or even as little as 1/16 inch, in America.

In some cases a small amount of inside lap is given, for the purpose of preventing the escape of the steam too early in the stroke.

In other cases inside lead is adopted, in order that the steam may have a freer escape. The former would be suitable for goods engines exerting large tractive force at slow speeds, and the latter for express engines working at a high speed of piston. The amount of either is rarely over ⅛ inch at the most.

With a valve worked by a single eccentric only, the suppression and release of the steam will always take place in one fixed manner, depending on the proportions of the valve and the throw of the eccentric. With equal lap and lead on both ends of the valve, these will occur at equal intervals in the advance of the crank from the centre line on both strokes. The positions of the piston, owing to the effect of the connecting rod being farther advanced at these times on one stroke than on the other, unequal amounts of steam will be received in the two cases, and unequal power given out. With a single eccentric valve gear this can only be prevented by making the lap greater on the forward end of the valve, or by lengthening the eccentric rod. Either of these changes will render the lead unequal on the two ends.

To obtain the greatly varying power required at different times in the locomotive with economy, it is necessary to be able to vary the time at which the steam shall be cut off from the cylinder. This may be done either by varying the *lead*, or the *lap*, or the *travel* of the valve.

The first of these methods accomplishes the object in a very imperfect manner, and to but a limited extent; for, as has been shown, the distance moved through by the piston before the cut-off occurs, bears the same relation to the length of the stroke as the lead of the valve to its total travel, and therefore it would not be possible, with ordinary proportions, to cut off much before quarter stroke, even were the eccentric set with 90° advance, or opposite to the crank. The great amount of pre-admission involved in the use of this means of effecting the cut-off and the early exhaust, make it quite unfit for practical use.

The second means produces a much better degree of expansion, but is not easily applied in practice; and the third, therefore, which gives a result very similar to the second, is the one universally adopted. This is effected now in by far the great majority of cases by the link motion in one of its three forms—viz., the shifting, stationary and straight link motions, according to the nature of the link employed in each case.

These consist merely of two eccentrics, set so that one shall be in position for forward and the other for backward motion, the ends of the eccentric rods being attached to a link so formed that the valve spindle may be made to receive its motion from any part of it.

The travel of the valve is varied by causing it to receive its motion from a point nearer to or farther from the centre of the link, in which position itself the amount of travel will be but the sum of the lap and lead of the valve, and the time occupied in passing over the distance covered by the lap therefore forms a large percentage of the time of each revolution. The same effect is produced in Hensinger von Waldegg's valve gear by reducing the travel in a similar manner, and imparting the lead by a separate mechanism from the motion of the piston.

In the stationary and straight link motions the advance at which the eccentrics are set is less or greater than the sum of the lap and lead of the valve, according as the eccentric rods are connected so as to cross when the crank is on the back or forward centre, the amount of this difference being expressed by $\frac{b\,d'}{L}$, in which d' represents the distance from the centre line of the motion at which the valve-rod link is situated when in full gear, *supposing it to coincide with the eccentric-rod joint.*

In Hensinger von Waldegg's gear the eccentric is set with no advance, and therefore needs but to have the direction of its motion reversed to fit it for working in back gear. The necessary lead is imparted to the valve by a lever worked by the piston of the engine.

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* By MR. FRED. J. SLADE.

In the shifting and straight link motions the alteration in the angle of the eccentric rods, when the crank is on centre with different grades of expansion, causes the lead to vary. In the other forms of gear mentioned, the position of the link or the contrivance for imparting lead being the same under all circumstances, no variation in this element occurs. This has generally been considered as a point in favour of the latter forms; but it may well be doubted whether the small increase, at least that occurs with high expansions, with the straight link be not an advantage at the higher speeds usual when cutting off early in the stroke.

The amount of this variation in the lead between full and mid gear varies directly as the distance through which the link is moved in passing from one position to the other.

With the straight link it follows that the variation is less than with the shifting link, since the link itself only moves through a portion of the space between the position required for full gear and that for mid gear, the remainder being traversed by the valve-rod link.

It follows, again, that the less the distance between the ends of the eccentric rods the less will be the variation of lead.

The formula $\frac{b\,d}{L}$, which we have found to express the amount of this variation, also shows us that it will be less as the length of the eccentric rods is increased, the throw of the eccentrics diminished, and the advance increased. If the eccentric rods are connected to the link so as to cross when the eccentrics are turned towards the link, the lead will *diminish* by this amount as we come into the position of mid gear; and if they are connected so as to cross when the crank is on the opposite centre, the lead will *increase* in the same proportion. In this expression, d represents the distance of the centre of the link from the centre line when in full gear, and for the shifting link, therefore, will be equal to half the distance between the eccentric-rod joints.

With all these forms of link motion it is possible to arrange the parts so as to produce a distribution of the steam which shall be so timed as to accord with the motion of the piston as influenced by the connecting rod. All such modifications of the arrangement of the link motion depend for their effect on one or both of two causes:—*first*, causing the link to rise and fall at certain portions of its motion, thus altering the angle of the eccentric rods, and thereby the distance of their points of attachment on the link from the shaft; and, *secondly*, changing the position of the valve connection in the link, bringing it farther from the centre at one time than at another. A clear conception of the manner in which these two changes take place explains all the phenomena of the different arrangements of link motions. Vertical motion may be imparted to the link either by altering the position of the weigh-shaft, or by moving the suspended centre of the link horizontally; and slight variations in the motion of the link are also produced by moving this centre vertically.

The shorter the link the more sensitive is it to changes in the position of the weigh-bar. The shorter the suspending link also, the more sensitive is the valve gear to such a change. The most potent means for varying the distribution is by moving the centre at which the link is suspended horizontally.

In the case of the shifting link, where the motion may be transmitted to the valve by the intervention of a rock shaft, as in the American engines, a correct distribution is produced by attaching the eccentric rods to the link at points back of the centre line.

In cases where the motion is communicated directly to the valve, the effect of so connecting the eccentric rods is to aggravate the inequality already produced by the connecting rod; and to obtain an equal distribution in such a case, it becomes necessary to move the suspended centre also in the same direction.

The position of the valve connection in the link is varied either as a consequence of the vertical motion of the link or independently by altering the position of the centre from which it is suspended.

Unequal lap may, if desired, be used with the link motion as with a single eccentric gear, but it is generally unnecessary to do so, since a proper locating of the centres of the motion will produce a satisfactory distribution without such a resort, with its attendant evil of unequal lead. In the straight link motion the equality of the lead on both strokes will only be maintained when the reversing arms are proportioned to each other according to the formula $\frac{l}{L}\left(1+\sqrt{1+\frac{L}{L}}\right)$, this being the ratio that should exist between the arm moving the valve-rod link and that connected to the expansion link.

In this motion the longer the valve-rod link the less will be the increase of lead toward mid gear, provided this length be not obtained at the expense of that of the eccentric rods.

In Heusinger von Waldegg's gear the connecting rod itself produces the correspondence between the motion of the valve and that of the piston. This is most complete near mid gear, but may be much varied by the means employed for communicating the motion of the cross-head to the intermediate lever, as well as by changing the position of the centre of the link, as in the ordinary link motions.

The earliest point at which the steam can be cut off with either of the forms of valve motion varies with the amount of lead obtaining at mid gear, since this measures the distance through which the valve must travel before the port can be closed. The shortest cut-off, therefore, will be obtained with either the stationary link motion or Heusinger von Waldegg's gear, and the longest admission in mid gear will be had with the shifting link, unless, indeed, we connect the eccentric rods so as to cross when the crank is on the back centre, in which case these conditions will be reversed. This, however, is never done. The periods yielded by the straight link motion will be a mean between those given, by the two of which it is compounded. With the dimensions which we have assumed we have seen that, with the first two gears, the steam may be cut off at about 10 per cent. of the stroke; and as the release occurs at half stroke, the maximum expansion possible is fivefold. With the straight link the admission was for about 12½ per cent. of the stroke, allowing, therefore, an expansion of nearly fourfold.

The shifting link reaches its minimum admission when the steam is cut off at 16 per cent., and the maximum period of expansion, therefore, is little over threefold.

These percentages may, of course, be varied by assuming other dimensions for the travel and lead. Thus, in the case of the straight link, for which the full table of results is given, with 5 inches travel of the valve in full gear, and 3½ inches in mid gear, with but $\frac{1}{16}$ inch maximum lead, we have an admission of but 10 per cent. With any given form and dimensions of link motion we can easily calculate the shortest point of cut-off that can be attained, by finding, from the formula $\frac{b\,d}{L}$, the increase of lead in mid gear, adding this to the lead in full gear, and dividing by twice the sum plus twice the lap, or the total travel of the valve at that point.

In Heusinger von Waldegg's or Walschäerts' gear we apparently have a variable expansion effected with a single eccentric. In reality, however, we must consider it as involving the equivalent at least of two, since the motion derived from the piston is exactly similar to that which would be given by an eccentric set in line with the crank. Indeed, we may look upon it merely as a resolution of the motion of an eccentric, with lead into its two components, the one parallel with, and the other at right angles to the centre line in such a way that, as it is only the latter that needs to have its direction changed in reversing the engine, this may be accomplished independently of the other, and the new resultant so obtained shall be equivalent to a single eccentric, with lead set for the backward motion of the engine. The forms of valve gear that we have considered are not the only ones that have been applied to locomotives; but the use of other forms is so exceptional as to render it unnecessary to discuss them here. Though not yielding the very best and most economical distribution of the steam, experience has pronounced the link motion the very best form of valve motion *for a locomotive* at present known. More theoretically perfect gears are no doubt preferable in other kinds of engines, but for this, simplicity, facility for reversing, and freedom from liability to derangement are the primary requisites, and these conditions the link motion very well fulfils.

CHAPTER XIII.*

THE GENERAL PRINCIPLES OF COMBUSTION.

COMBUSTION is the chemical combination of any number of bodies, causing the generation of heat.

It has been a long-established mistake to consider the substances entering combustion in two different groups—viz., as *combustibles* or *fuel*, and as *supporters of combustion*, the latter being generally represented by oxygen, or atmospheric air. There is in reality no difference between the action of either combustible or supporter of combustion, and their relative functions may be reversed at will. A jet of hydrogen, for example, may be ignited in an atmosphere of oxygen, or in the open air, and it will produce a flame, the hydrogen appearing as combustible, and the oxygen (pure, or mixed with other gases in the air) representing the supporter of combustion: but it is equally feasible to force a jet of oxygen or air into a receiver filled with hydrogen; and that jet, when ignited, will burn in the atmosphere of hydrogen, the oxygen or air now being the fuel, and the hydrogen the supporter of combustion.

The different elements entering into chemical combination in the act of combustion are therefore upon an equal footing—they are all combustible, and all supporters of combustion. Considerations of this kind are not immaterial to the correct understanding of the principles upon which the generation of heat in the furnace of a locomotive is based; and the earlier propagation of these views would have saved a considerable amount of misspent ingenuity and wasted labour upon many worthless schemes for supposed economy of fuel which have been brought forward from time to time.

The principal elements which enter combustion in the locomotive fire box are carbon, hydrogen, and oxygen.

Carbon is the principal, and almost the only element contained in coke. It is usually mixed with a small percentage of mineral matter, which forms the ashes after complete combustion. The amount of ashes varies with different kinds of coke, and ranges between 1 and 10 per cent. of the total weight of coke.

The combination of carbon with oxygen produces two different gases, according to the proportion of the two bodies combining. *One atom of carbon* combined with *one atom of oxygen* produce *carbonic oxide*—a gas capable of further combustion in combination with another atom of oxygen. This further and complete combustion produces *carbonic acid*, containing *one atom of carbon* and *two atoms of oxygen*.

The atomic weight of carbon is = 6, that of oxygen = 8—*i. e.*, 6 parts by weight of carbon and 8 parts of oxygen form carbonic oxide. Six parts of carbon and 16 parts of oxygen produce carbonic acid. The complete combustion of 1 lb. of carbon requires $\frac{16}{6} = 2\frac{2}{3}$ lbs. of oxygen.

Hydrogen does not appear in the pure state in any part of the locomotive furnace, but it forms an important and valuable component of the gases contained in the coal, and also of the substance of wood. During combustion, the hydrogen is separated from the carbon of the coal-gas, and each of the elements combines with the oxygen of the air, one atom of hydrogen requiring one atom of oxygen for the formation of water. Hydrogen having an atomic weight equal to one; 1 lb. of hydrogen takes 8 lbs. of oxygen for its combustion. It will be seen from the above that 1 lb. of hydrogen takes three times the quantity of oxygen for its combustion that 1 lb. of pure carbon does.

The quantity of heat produced by different substances in combustion has been subjected to scientific investigations and experimental researches. Count Rumford found, by direct experiment, that the heating power of 1 lb. of carbon is about 14,000 units—*i. e.*, by the complete combustion of 1 lb. of carbon and $2\frac{2}{3}$ lbs. of oxygen a quantity of heat is generated which will raise the temperature of 14,000 lbs. of water one degree Fahr. The heating power of hydrogen has been established in a similar manner, and it amounts to 42,000 units, or three times the heating power of carbon.

It appears in this instance that the heating powers of carbon and hydrogen bear the same proportion to each other as the quantities of oxygen these two bodies absorb in combustion.

This is a very important fact, allowing of a general application to all kinds of fuel used in the locomotive—viz., *The heating power of any kind of fuel is proportionate to the quantity of oxygen absorbed by it in complete combustion.* The combustion of 1 lb. of oxygen produces about 5,250 units of heat. This quantity remains constant, whether the oxygen be combined with carbon, hydrogen, or any combination of both elements.

Oxygen also forms a component part of some kinds of fuel, as, for instance, wood, peat, and some kinds of coal. In this combined state the oxygen does not add to the heating power of the combustible—on the contrary, it diminishes it. The explanation of this will be found by considering that every atom of oxygen contained in the fuel exists in combination with one atom of hydrogen in the form of water, and that therefore neither the one nor the other atom can enter any new combination, and both are unavailable for combustion. To find the heating power of a substance composed of carbon, hydrogen, and oxygen, we have *to subtract 1 part by weight of hydrogen for every 8 parts of oxygen it contains, and to determine the heating power of the rest.*

If, for example, the heating power of dry wood is to be found, we find, on reference to analysis, that dry wood of many trees contains—

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* By MR. FERDINAND KOHN.

Carbon, . 40 per cent.
Hydrogen, $6\frac{2}{3}$ „ } = 60 per cent. of water.
Oxygen, . $53\frac{1}{3}$ „ }
100

In this instance the hydrogen to be balanced by $53\frac{1}{3}$ parts of oxygen is $\frac{53\frac{1}{3}}{8} = 6\frac{2}{3}$ parts, or exactly the total quantity present. There remains, therefore, only 40 parts of carbon available for combustion. One pound of dry wood, therefore, has 40 per cent. of the heating effect of 1 lb. of pure carbon—*i. e.*, $\frac{14,000 \times 40}{100} = 5,600$ units of heat.

The products of combustion in the locomotive fire box are all gaseous. The carbon is converted into carbonic acid or carbonic oxide; the hydrogen into water, which exists as steam at that high temperature; and the unconsumed oxygen, together with the other gases from the atmosphere—viz., nitrogen, carbonic acid, &c.—are the other components of the gaseous current passing through the tubes of the boiler. It is obvious that the furnace gases are the immediate recipients of the total heat generated by combustion, and from their quantity and specific heat will depend the temperature in the fire box.

The temperature produced by a certain kind of fuel is therefore dependent on the chemical elements it contains. Suppose 1 lb. of pure carbon to be burnt in pure oxygen, its total heating effect of 14,000 units would have to be taken up by the carbonic acid formed—viz., $3\frac{2}{3}$ lbs. If the specific heat of carbonic acid were equal to that of water, these 14,000 units would raise $3\frac{2}{3}$ lbs. of gas to, say, $\frac{14,000}{3\frac{2}{3}} = 3,111$ degrees; but the specific heat is only 0·221 of that of water, and the rise of temperature will therefore be $\frac{3,111}{0·221} = 14,083$ degrees, or $\frac{14,000}{3\frac{2}{3} \times 0·221}$.

To find the temperature produced by any kind of fuel, we have to divide the total heat due to its combustion by the total weight multiplied by the specific heat of the products of combustion.

It is obvious that in case of there being several different gases of different specific heats produced by the combustion, we must divide *by the sum of the products of their respective quantities, each quantity being multiplied by the specific heat of the respective gas.* See following table.

Combustion of Coke.—To find the temperature produced by the complete combustion of coke in the locomotive furnace, we take the total heat produced by 1 lb. of coke = 14,000 units. For this combustion $2\frac{2}{3}$ lbs. of oxygen are required, and with these are associated $9\frac{1}{3}$ lbs. of nitrogen in the atmospheric air, making a total of 12 lbs. The products of combustion, therefore, will weigh 13 lbs., being $3\frac{2}{3}$ lbs. of carbonic acid and $9\frac{1}{3}$ lbs. of nitrogen. The specific heat of nitrogen is 0·275, that of carbonic acid, as above, 0·221. The temperature will be = $\frac{14,000}{3\frac{2}{3} \times 0·221 + 9\frac{1}{3} \times 0·275} = \frac{14,000}{0·811 \times 2·566} = \frac{14,000}{3·377} =$ 4,734 degrees.

TABLE SHOWING THE SPECIFIC HEAT AND OTHER PROPERTIES OF WATER AND GASES.*

Name.	Specific Heat.	Increase of Temperature of One Pound by One Unit of Heat.	Specific Gravity at 62°.	Cubic Feet in One Pound at 62°.	Increase of Temperature of One Cubic Foot of Gas, measured at 62°, per Unit of Heat.
		Degrees.			Degrees.
Water, . . .	1·0000	1·	816·000	0·016	...
Air, ...	0·2669	3·75	1·000	13·0	48·75
Carbonic Acid, .	0·2210	4·52	1·527	8·5	38·50
Carbonic Oxide,	0·2884	3·47	0·967	13·5	46·80
Nitrogen, . .	0·2754	3·63	0·969	13·1	48·50
Oxygen, . . .	0·2361	4·23	1·111	11·7	49·50
Hydrogen, . .	3·2936	0·30	0·070	186·0	55·80
Steam, . . .	0·8470	...	...	...	...

The temperature in a locomotive fire box burning coke would be 4,734 degrees as a maximum. This is never obtained in practice, since a surplus of atmospheric air is required for the proper combustion, this surplus being estimated at 3 or 4 lbs. of air per pound of fuel; besides this, the cooling effect of the sides and crown of the fire box must be taken into consideration. It is probable, however, that in the hottest part of the fire box a temperature of above 3,000 degrees is maintained with good coke as fuel.

In a similar manner it can be ascertained by calculation that the temperature produced by the combustion of coal (however perfect and economical such combustion may be effected) can never be as high as that produced by coke. The theoretical maximum of temperature for Newcastle coal is only about 3,800 degrees, for Welsh coal about 4,000, and for Anthracite 4,500. Allowing for surplus of air and cooling effects, we must estimate the temperature in the fire box of a coal-burning locomotive about 20 per cent. lower than that obtained by coke. With wood as fuel, the temperature in the fire box can never reach much more than 2,000 degrees, and it is probably below this in most cases. The quantity and quality of the fuel to be burnt in a given time in the fire box determines the quantity of the oxygen and (by the constant proportion of oxygen and nitrogen in the atmosphere) the exact quantity of air to be admitted, also, to some extent, the manner in which such admission of air is to take place.

The weight of 1 cubic foot of air at 60 degrees is 0·0766 lbs. For the combustion of 1 lb. of carbon 12 lbs. of air are required, the volume of which is—

$$12 \times 0·0766 = 160 \text{ cubic feet.}$$

With the addition of the necessary surplus, the quantity of air which must pass through the grate for the combustion of 1 lb. of carbon, or pure coke, will reach 200 cubic feet.

The gaseous products resulting from the complete combustion of carbon or coke occupy the same volume (at equal temperatures) as the air supplied to the fuel, since carbonic acid, owing to its great specific weight, has the same volume as the oxygen which it contains. At a temperature of 3,000 degrees the volume of these gases is 1,340 cubic feet; but this is speedily reduced by the cooling effect of the fire box and tubes, and the gases leave the smoke box at a temperature of about 600 degrees. Their volume is about 400 cubic feet, or double the volume of air which enters the fire box.

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* From D. K. Clark's *Railway Machinery*, p. 122.

The furnace gases escaping at that elevated temperature carry a certain amount of heat with them, which is lost so far as transmission to the boiler is concerned.

To find the quantity of heat contained in the escaping gases at 600 degrees, we have to consider their relative quantities and specific heats. These gases consist of

3⅔ lbs. Carbonic Acid, specific heat = 0·221.
9¼ „ Nitrogen, . „ = 0·275.
3 „ Air, ... „ = 0·266.

At 600 degrees these gases contain about 540 degrees of extra heat compared with the air at 60. This amounts for

Carbonic Acid, 3⅔ × 0·221 × 540 = 437 units.
Nitrogen, . 9¼ × 0·275 × 540 = 1,496 „
Air, . . . 3 × 0·266 × 540 = 432 „
Total, . . . = 2,365 units.

This loss amounts to 16¼ per cent. of the total heat generated. To compare the results of these considerations with actual practice, it is most convenient to find the quantity of water at 60 degrees which is evaporated by the combustion of 1 lb. of coke. Taking the total heat contained in 1 lb. of steam at 320 degrees or 90 lbs. pressure = 1,211 units, we find the theoretical evaporative power of 1 lb. of coke $= \frac{14{,}000}{1{,}211 - 60} = 12{\cdot}16$ lbs. of water.

From this the quantity of 2,365 units being carried off by the furnace gases a deduction is to be made—viz., $\frac{2{,}365}{1{,}211 - 60} = 2$ lbs. of water; the remainder being thus 10·16 lbs.

The utmost practical performance of 1 lb. of coke in locomotive boilers has been found equal to the evaporation of about 9½ lbs. of water, which would show the following disposition of the total heat generated:—

Formation of Steam, . .	10,920 units =	78 per cent.
Loss by the Heat of gases,	2,365 „ =	16¼ „
Loss by Ashes and accidental causes, . . .	715 „ =	5¼ „
	14,000 units.	100 parts.

Coke being considered as pure carbon, admits of a very simple investigation with regard to the manner and results of its combustion. In modern locomotive practice, however, the use of coal as a fuel has become so general that a closer consideration of the somewhat more complicated process of its combustion will be necessary.

Combustion of Coal.—The chemical elements contained in some kinds of British coal are shown in the following table:—

COMPOSITION OF COALS.

Name.	Specific Gravity.	Carbon.	Hydrogen.	Nitrogen.	Sulphur.	Oxygen.	Ash.	Per Centage of Coke left by each Coal.
	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.
Anthracite, ...	1·375	91·44	3·46	0·21	0·79	2·58	1·52	92·9
Graigola, ...	1·300	84·87	3·84	0·41	0·45	7·19	3·24	85·5
Oldcastle's Fiery Vein,	1·289	87·68	4·89	1·31	0·49	3·39	2·64	79·8
Dalkeith Jewel Seam,	1·277	74·55	5·14	0·10	0·33	15·51	4·37	49·8
Grangemouth, . . .	1·290	79·85	5·28	1·35	1·42	8·58	3·52	56·6
Broomhill, ...	1·250	81·70	6·17	1·84	2·85	4·37	3·07	59·2
Park End, Lydney, .	1·283	73·52	5·69	2·04	2·27	6·48	10·00	57·8

This analysis shows the relative quantities of the three elements of vital importance in combustion—viz., carbon, hydrogen, and oxygen, and thereby furnishes the necessary data for investigation.

Selecting Grangemouth coal, on account of its comparatively high per centage of hydrogen, as one of the most characteristic specimens, we must first of all make the necessary deduction for the water which we consider represented by the oxygen in the coal.

One hundred pounds of Grangemouth coal contain:—

$$\left.\begin{array}{lr} \text{Carbon, ...} & 79{\cdot}85 \text{ lbs.} \\ \text{Hydrogen, . .} & 5{\cdot}28 \text{ „} \\ \text{Oxygen, . . .} & 8{\cdot}58 \text{ „} \end{array}\right\} = \left\{\begin{array}{lr} \text{Carbon, . .} & 79{\cdot}85 \text{ lbs.} \\ \text{Hydrogen, .} & 4{\cdot}21 \text{ „} \\ \text{Water, . .} & 9{\cdot}65 \text{ „} \end{array}\right.$$

Having in this manner made allowance for the oxygen, we obtain for the combustion of

79·85 lbs. of Carbon × 14,000 units = 1,017,900 units.
4·21 „ Hydrogen × 42,000 „ = 176,820 „
Total, . . . = 1,194,720 units.

The Oxygen required for the Carbon is = 212·90 lbs.
„ „ „ Hydrogen is = 33·68 „
Total Oxygen consumed, = 246·58 lbs.
Associated Nitrogen from Air, ... = 821·93 „
Total Air consumed, . = 1,068·51 lbs.
Surplus of Air in fire box, 25 per cent, = 267·15 „
Total Air admitted, . = 1,335·66 lbs.

The products of combustion are,

Carbonic Acid, 29·275 lbs. . × 0·221, specific heat = 64·698

$$\text{Steam,}\left\{\begin{array}{l} \text{Water originally contained, ... } = 9{\cdot}65 \\ \text{Water formed by the free Hydrogen, . } = 37{\cdot}89 \end{array}\right\} 47{\cdot}54 \times 0{\cdot}847 = 40{\cdot}275$$

Nitrogen, ... 821·93 lbs. = 0·275 = 226·230
Surplus Air, ... 267·15 „ = 0·266 = 71·061
Total, ... = 402·264

These products of combustion, escaping at a temperature of 540 degrees above that of the atmosphere, carry off 540 × 402·264 = 217,222·56 units.

The disposition of the heat produced by the combustion of 100 lbs. of Grangemouth coal is as follows:—

Total Heat, ... 1,194,720 units.
Loss by Heat of furnace gases escaping, 217,222 „
Rest, ... 987,498 units.

The total available evaporative power of 100 lbs. of coal is therefore $= \frac{987,498}{1,211·54 - 60} =$ 858 lbs. of water; or for 1 lb. of coal = 8·58 lbs. of water at 60 degrees.

The following table shows the results of actual experiments with the qualities of coal described in the preceding table:—

Name.	Constituent Carbon.	Constituent Hydrogen.	Coke yielded by destructive Distillation.	Water actually evaporated from 212° by 1 lb. of Coal.
	Per Cent.	Per Cent.	Per Cent.	Lbs.
Anthracite, . .	91·44	3·46	92·9	9·46
Graigola, . . .	84·87	3·84	85·5	9·35
Oldcastle's Fiery Vein,	87·68	4·89	79·8	8·94
Dalkeith Jewel Seam, . . .	74·55	5·14	49·8	7·08
Grangemouth, .	79·85	5·28	56·6	7·40
Broomhill, . .	81·70	6·17	59·2	7·30
Park End, Lydney,	73·52	5·69	57·8	8·52

The quantity of water actually evaporated by 1 lb. of Grangemouth coal is = 7·40 lbs.

Thus the whole heat of combustion of 1 lb. of coal is disposed of as follows:—

7,400 units = 62·2 per cent. in the formation of Steam.
2,172 „ = 18·3 „ Loss by the Heat of gases in the smoke box.
2,375 „ = 19·5 „ Waste.
11,947 „ = 100 parts.

It has been a matter of astonishment to many writers on this subject that the evaporative power of coal did not considerably increase with its per centage of hydrogen, since it had been inferred, from the great heating power of hydrogen gas, that the coal containing more gas would be stronger than coal that contained less hydrogen.

Mr. D. K. Clark, in his work on *Railway Machinery*, endeavoured to explain the inferiority of coal to coke as a fuel, by the fact that the preparatory distillation of the combustible gases of coal, before combustion, involves an outlay of heat for their conversion into gaseous form, which may go some length to balance the heat arising from their combustion; also, that oxygen combines with twice its volume of hydrogen to form aqueous vapour, of which the volume is equal to that of the elementary hydrogen as a gas, the product of combustion having double the volume of the oxygen employed, and that it is this excess of volume to which the absorption of the heat is due. Carbonic acid, on the contrary, possesses the volume simply of its constituent oxygen: there is no extension of volume for the conversion of carbon into gaseous form, and there is no absorption of heat thereby. Mr. Clark does not seem to have been entirely satisfied with this explanation, as he remarks,—"The distinction here made appears to explain the practical neutrality of solid hydrogen as a dispenser of heat by combustion;" and further, he adds,—"The inferiority of coal is due, not wholly to its composition, but also to the unfavourable conditions under which it is burnt." The volume of the gaseous products of combustion has no influence upon the quantity of heat absorbed by them. The weights and temperatures being given, the more voluminous gas is in the same proportion less in specific weight; nor is the quantity of heat absorbed in the first distribution of the coal gas sufficient to balance a considerable portion of the heat due to the combustion of hydrogen. The main cause for the small heating effect of the hydrogen in the coal is due to the great specific heat of the steam produced by its combustion. From the table of specific heats it will be seen that 1 lb. of steam requires 0·8470 units of heat for each degree of temperature, while carbonic acid only takes 0·221 units for each degree, or not much more than one quarter of the former. One pound of steam escaping into the smoke box carries with it nearly as much heat as 4 lbs. of carbonic acid. Supposing, therefore, in accordance with general practice, that the gases of combustion derived from coal escape from the smoke box at the same temperature as those from coke, the necessary loss, owing to the great quantity of heat that is contained in the aqueous vapour, will be the most important drawback upon the evaporative duty of coal.

This leads to the conclusion that, with coal-burning locomotives, there is a greater margin left for economy by the reduction of the temperature of the smoke-box gases than with locomotives using coke as their fuel.

The combustion of coal in the locomotive fire box requires a peculiar method of admitting the air, in order to effect perfect combustion. When fresh coal is charged into a furnace, the first result is the liberation of its gaseous components, in the shape of hydrocarbon gas, consisting of 6 parts by weight of carbon, and 1 part of hydrogen. This gas, when properly mixed with air, burns with a clear, bright flame, of great heating power, *though not of so high a temperature as is produced by the combustion of coke*. It is of the utmost importance that these gases should be intermixed with a sufficient quantity of air at the moment of their generation, and close to the spot where they are generated, otherwise the rapid current would draw them through the tubes of the boiler in their unburnt state, and their total heating power would be wasted. The air for coal fires is therefore in most cases admitted into the fire box in two separate ways—viz., partly through the grate, and partly, by special contrivances, direct into the fire box above the coal, so as to mix with the coal gases rising from the fuel. The quantity of air to be admitted either way depends upon the relative proportion of gases contained in the coal.

With the specimen considered above, the quantity of hydrocarbon gas produced by 100 lbs. of coal would be—

4·21 lbs. Hydrogen × 25·26 Carbon . = 29·47 lbs.
Requiring Oxygen: for the Hydrogen. = 33·68 „
„ Carbon . . = 67·36 „
Total, ... = 101·04 lbs.
Quantity of Air corresponding, ... = 454·66 „
Surplus of Air, 25 per cent., ... = 113·45 „
Quantity of Air to be admitted above grate = 568·11 lbs.
Do. do. do. through grate = 767·55 „

Reducing these quantities to cubic feet of air, at 60 degrees per pound of coal, we find—

Air above grate, 75 cubic feet, } per pound of coal.
" through grate, 101 " }
Total, . . 176 cubic feet.

The various methods of admitting air to the gaseous fuel above the grate have been described in the historical section, and will find further consideration in another chapter. The present investigation deals with the subject without reference to any particular arrangement for coal-burning. Whatever the method for admitting the air may be, it will be observed that the exact quantity which shall pass over and through the grate is fixed by the composition of the coal. From this it follows that arrangements which give excellent results with one particular kind of coal cannot, without alterations in the proportions and sizes of their principal parts, be successfully applied to other qualities of coal differing from the first in their chemical composition. Even with the same kind of coal there may be various stages in the process of combustion, presenting different conditions to the air admitted to the fire box. If, for instance, a great quantity of fresh coal be thrown upon the grate in a mass, the immediate result would be the disengagement of so great a volume of gas that the regular supply of air provided for the average consumption above the grate would be insufficient for complete combustion; while, on the other hand, the cooling effect upon the strata of incandescent fuel upon the grate would prevent the complete combustion of the solid carbon by the air which passes the grate bars, and would thus tend to increase the quantity of incompletely-burnt gases by filling the fire box with carbonic oxide. After the disengagement of the greater bulk of the gases, on the contrary, there would be a surplus of air above the grate which could not be made use of for combustion, and which would lower the temperature of the fire box and tubes, carrying off a quantity of wasted heat to the smoke box.

The only practical method of counterbalancing the effects of this irregular combustion of coal is skilful firing. The aim of the stoker must be to have a proportionate mixture of coal in all stages of combustion spread over the grate, and thereby to approach as nearly as possible the average constant proportions which are assumed by theory. This being attended to, the proportions of air passages once established for each kind of coal will remain sufficiently constant for all practical purposes.

The admission of the proper and exact quantity of air through and above the grate forms the whole art of coal-burning in locomotive engines. Insufficiency of air supply causes the escape of dense black smoke; and this having been the great and striking fault of the first locomotives fired with coal, it has been considered successful practice if any locomotive could burn coal without producing visible smoke. It is obvious, however, that the mere fact of so-called smoke-burning must not of necessity be combined with the maximum of evaporation duly obtained from the coal, and is therefore in itself no criterion for the success of locomotive engineering. The disappearance of all visible smoke may be due to a considerable surplus of air over and above the proper quantity required. The combustion in such a case would be perfect; but the products of combustion, being unnecessarily augmented by the useless ballast of the surplus air, would reduce the temperature of the flame, and lessen its heating effect, by carrying off a quantity of heat escaping through the chimney at a high temperature.

To compare the relative economy of using raw coal as fuel in the locomotive, or converting it into coke before using it, we find from the table that 100 lbs. of Grangemouth coal will produce 56·6 lbs. of coke, capable of evaporating 9·5 × 56·6 = 537·7 lbs. of water at 60 degrees. The direct combustion of 100 lbs. of coal would evaporate 660 lbs. of water from the same original temperature. With perfect combustion of coal, the economy of coal-burning is therefore very considerable.

Combustion of Wood.—The evaporative power of dry pine wood is generally considered to be = 2½ lbs. of water per pound of wood. This fixes the equivalent for 1 lb. of coke at about 4 lbs. of wood, and for 1 lb. of coal at 2½ lbs. of pine. The weight of 1 cubic foot of dry pine is about 21 lbs.

The practical conclusions to be drawn from the preceding investigation of the combustion in a locomotive fire box are as follows:—

1. Successful practice requires complete combustion of the carbon and hydrogen available in the fuel.

2. To find the quantity of free carbon and hydrogen, it is necessary to deduct 1 part by weight of hydrogen, or 6 parts of carbon from the total contents of the fuel for every 8 parts of oxygen contained in the same.

3. One pound of coke requires about 200 cubic feet of air for its combustion: the air may be admitted through the grate only.

4. One pound of coke is capable of evaporating 9·5 lbs. of water at 60 degrees; in common practice, its evaporative power is 8½ lbs. of water.

5. The temperature produced by the combustion of coke, in the hottest part of the fire box, may be estimated at 3,000 degrees.

6. The gases produced by the combustion of coke carry 16½ per cent. of the total heat generated into the smoke box, which they leave at a temperature of 600 degrees.

7. The complete combustion of coal requires the admission of air both through and above the grate; the relative proportion and the total quantity of air admitted in both ways depends upon the per centage of gaseous components in the coal.

8. Insufficient admission of air causes smoke and the loss of heating effect by incomplete combustion. A surplus of air reduces the temperature of the flame, and causes waste of heat.

9. The evaporative duty of coal per pound weight averages about 6 lbs. of water; in regular practice, the maximum being about 8 lbs.

10. The temperature produced by coal in the fire box is lower than that obtained from coke.

11. The products of combustion from coal have a higher specific heat than those from coke: they carry off a quantity of heat equal to 18·3 per cent. of the total heat produced. This per centage varies with the contents of hydrogen in the coal.

12. The combustion of coal must be made as nearly uniform as possible by skilful firing.

13. Coal, when completely and properly burned in the locomotive, affords a great economy, as compared with the coke produced from the same.

14. The evaporative power of dry pine wood is, in practice, 2½ lbs. of water, the maximum having been found at 4 lbs. One pound of coke is equivalent to 3 lbs. of wood; 1 lb. of coal is equivalent to 2¼ lbs. of wood; 1 cubic foot of wood weighs 21 lbs.

15. The temperature produced by wood is generally less than 2,000 degrees.

CHAPTER XIV.*

THE FUNCTIONS OF THE LOCOMOTIVE BOILER.

THE boiler has the function to transmit the heat generated in the fire box to the water and steam contained within it, and to form a reservoir for the steam produced.

The quantity of heat which passes through a solid body, when in contact with a hotter medium on one side and with a colder one on the other, depends—1. On the extent of surface in contact with both bodies; 2. On the difference of temperature existing on the two sides of the solid body; 3. On the time during which the transmission of heat is continued; and, 4. On the specific conducting or heat-transmitting power of the solid body in question.

With regard to the area in contact, it is obvious that the quantity of heat transmitted will be in direct proportion to the extent of heat-transmitting surface. The area which is in direct contact with the hot gases of the furnace on one side, and with the water contained in the boiler on the other side, has been termed the *heating surface*, and has become the first and most important element for measuring the evaporative power of a boiler.

Heating Surface.—The importance of the extent of heating surface has been very often misunderstood. Engineers sometimes are tempted to rely too much on the absolute number of square feet of heating surface in locomotive boilers; yet nothing can be more deceptive than the supposed heat-transmitting power of a square foot of internal surface.

Mr. Armstrong found that a cubical metallic box, submerged in water and heated from within, generated steam from its upper surface more than twice as fast, per unit of area, as it did from the sides when vertical, and that the bottom yielded none at all. These remarkable differences are owing to the difficulty with which steam separates from a vertical surface to give place to fresh charges of water, and to the impossibility of leaving the inverted surface at all. By slightly inclining the box, the elevated side much more easily parted with the steam, and the rate of evaporation was increased; while on the depressed side the steam hung so sluggishly as to lead to an over-heating of the metal.† This fact, of an inclined surface producing more steam than a vertical one, has induced many authorities in locomotive engineering to advocate the inward inclination of the fire-box sides. It is doubtful, however, whether anything is gained in this respect, since this inclination of the sides reduces the area of the furnace crown, so that the gain may be counterbalanced by a proportionate loss in that point. The advantage of inclined fire-box sides seems principally to consist in the better circulation which is thereby provided for in the water-space between the fire box and the shell of the boiler.

Still more than in the fire box, the importance of other elements besides the mere extent of heating surface is shown, by experience, with the arrangement of the tubes in locomotive boilers. It has been proved by repeated experiments that great numbers of tubes crammed into a locomotive boiler, without due regard to facility of circulation, produce an inferior evaporative effect than a smaller total heating surface afforded by a more limited number of tubes judiciously arranged.

On the other hand, the position of the heating surface has been much overrated by several great authorities. Experiments with locomotives of various proportions seemed to show a close connection between the total evaporative power of a locomotive boiler and the extent of its fire-box surface.

From these experiments it seemed to appear that the evaporative power of a locomotive boiler is independent of the tube surface, and simply follows the rate of 2 cubic feet of water evaporated per square foot of fire-box surface.‡

It has been, however, proved by theory and by practice that this is a fallacy, and that tube surface is fully as important for evaporation as that of the fire box.

Temperature of Furnace Gases.—The heat transmitted by a solid body from a hotter medium to a colder one is in direct proportion to the difference of the respective temperatures of the two. The evaporation with a given heating surface will therefore be increased by a higher temperature of the furnace gases. This gives a decided advantage to coke as a fuel for locomotives, the temperature of its products of combustion being considerably higher than that produced by coal. The temperature of the steam in the boiler must be considered to represent the lower temperature; and from this it would appear that higher pressure of steam, which is equivalent to higher temperature, would cause an inferior evaporative effect. This, however, is not the case in practice, since the gases always escape at a considerably higher temperature than that of the steam in the boiler, the pressure of which has therefore no sensible influence upon the rate of transmission of heat.

Time of Contact.—The time during which the transmission of heat takes place is the third element to which the quantity of heat transmitted is directly proportioned, provided that the temperature of the heating and of the heated substance remain constant during the process. In the tubes of a locomotive boiler the temperature of the gases does not remain constant; extension of time is therefore of less importance, and is often sacrificed in order to obtain more important advantages. Slow motion of the furnace gases in the tubes is obtained by making the total flue-way or sectional area of tubes as large as possible, the speed of the current being inversely proportioned to the total area of air passages.

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* By MR. FERDINAND KOHN.

† Tredgold *On the Steam Engine*, vol. i.

‡ *Gauge Commissioners' Report*, 1846.

It is at the same time to be remarked, that the mere provision of ample flue-way is in itself no more efficient than the absolute size of heating surface. The flues must be so arranged that the gases will pass through all in equal proportion. The products of combustion being at a high temperature, are found to take the nearest or shortest course to the chimney, regardless of whatever surface may have been elsewhere provided. It is therefore an important object for consideration to arrange the grate and fire box in such a manner as to give each row of tubes its fair share of the gaseous current. Under ordinary circumstances the distance from the back of the fire box to the tubes is greater than that from the front part of the grate, so that the great bulk of the air would be drawn in at the front end of the grate, and pass through the lowest rows of tubes; this being the shortest way, and offering the least resistance. The upper rows of tubes and the parts of the grate near the fire-door plate would in this case lose part of their efficiency. To counteract this influence the grate is sometimes inclined towards the front of the fire box, which partly equalizes the distances from the tube plate, and also has the effect of collecting a thicker layer of fuel at the front of the grate, which offers more resistance to the passage of air on its shortest course, and thereby equalizes the admission of air all over the grate.

Specific Heat-transmitting Power.—To consider the specific heat-transmitting power of a boiler plate, we have to distinguish three different elementary functions, viz.—1. The reception of heat at the surface exposed to the source of heat; 2. The conduction of heat within the plate; and 3. The emission of heat by the surface in contact with water.

Let A be the quantity of heat absorbed by one square inch of boiler plate in one minute, when exposed to the furnace gases of a locomotive at a given temperature.

B, the quantity of heat transmitted within the boiler plate, under the same circumstances, through *the thickness of one inch.*

C, the quantity of heat given by one square inch of the same plate to the water in contact with it.

Then we may call A the unit of absorbing power.
B, the unit of conducting power.
C, the unit of emitting power.

For a plate of the thickness = t, the figures A and C remain unaltered (all circumstances being the same); but the figure B will have to be divided by t, since the quantity of heat conducted is inversely proportioned to the distance through which it has to travel—*i. e.*, the thickness of plate.

Now, if the heat within the fire box be constant, as well as the temperature of the water in the boiler, the plate will, after a certain time, come to a stationary temperature, at which all the heat absorbed by it will be passed through and given to the water, the plate neither gaining nor losing heat. At this moment the three figures

$$A = \frac{B}{t} = C = H$$

must be equal to each other, and equal to the total heat transmitted.

In order to find the relation which all the three elements bear to the total heat, the above equation may be expressed as follows :—

$$\frac{1}{A} = \frac{t}{B} = \frac{1}{C} = \frac{1}{H}, \text{ or}$$

$$\frac{1}{A} + \frac{t}{B} + \frac{1}{C} = \frac{3}{H}, \text{ from which}$$

$$H = \frac{3}{\frac{1}{A} + \frac{t}{B} + \frac{1}{C}} *$$

This expression shows that the heat-transmitting power of a boiler plate increases with the heat-absorbing, conducting, and emitting power, and decreases with its thickness; *but the heat-transmitting power is not inversely proportioned to the thickness of the plate.*

The influence of the thickness appears all the less important the larger B and the smaller A and C are made—*i. e.*, with good conducting power of the metal, and with inferior heat-transmitting capabilities of its surfaces, the thickness of plate becomes of less importance for the transmission of heat.

In practice, this inferiority of the surface is the general rule. The boiler plates are coated with a certain thickness of oxide, and with a layer of soot on one side, and with oxide and scale on the other side, these superposed layers of inferior conductors of heat having the effect of lessening the heat-absorbing and emitting power of the surfaces. From this it will appear that within certain limits the thickness of plates in a locomotive boiler is of small influence upon the heat-transmitting or evaporating quality of the boiler. With clear surfaces, however, thin boiler plates give a better evaporative effect than thicker plates.

The thickness of plate has, however, another important influence upon the transmission of heat. The conduction within the body of the plate being uniform throughout, it follows that the plate will have the highest temperature at the surface exposed to the furnace gases, and that the temperature will gradually diminish within the plate until it reaches its lowest point at the surface in contact with the water. Now, the difference between the temperature of both surfaces of the plate will be in direct proportion to its thickness. Supposing, therefore, the temperature of the surface in contact with the water to be given, it follows that the surface exposed to the fire will become more and more heated as the thickness of the plate is increased. The liability of thick boiler plates to injury from excessive heating is well known, and it forms a more important objection to excessive thickness of plate than the inferior evaporative power, which latter very often has been much overrated.

The absolute value of the figures A, B, C, representing units of heat-transmission under varied circumstances, have never been established. It would require a series of carefully conducted experiments with plates of different materials, and with varied conditions of their surfaces, to supply the necessary data.

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* This equation has the same form as those more general formulæ which have been found by Fourier, Poisson, and Redtenbacher.

Transmission of Heat with Decreasing Temperature of the Furnace Gases.—In the preceding deductions it has been assumed, for the sake of simplicity, that during the time of contact between the heating surface and the furnace gases the temperature of the latter remain constant.

This condition practically exists only with regard to very small areas of boiler plate; while, in all considerations of the heating surface of a boiler as a whole, or of any more extended part of the same, the transmission of heat must be considered under the condition of a decreasing temperature of the furnace gases when passing over the heating surface.

To approach the conditions of practice as closely as possible, suppose Fig. 195 to represent a locomotive boiler tube surrounded with water, and heated by the current of furnace gases passing through the tube.

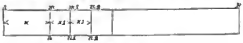

Fig. 195.

Take the cross section of the tube at any particular place, and suppose a very small length cut out, same as represented in the figure, by the piece $m\ m_1\ n\ n_1$. Call G = the quantity of gases in lbs. passing through the section $m\ n$ per minute, t = the temperature of those gases at $m\ n$ above that of the water surrounding the tube. The temperature, t, will depend on the distance, x, from the end of the tube at which $m\ n$ is situated. After passing through the small length $m\ m_1$, the gases have lost a certain quantity of heat, which, as has been shown already, is proportionate to the length of $m\ m_1 = x_1$, and to the difference of temperature between gases and water. Take the heat which would be absorbed per minute by 1 inch length of this tube from gases one degree hotter than the surrounding water as unit of absorption: then the heat absorbed by the small piece $m\ m_1\ n\ n_1$ is: $H_1 = x_1, t$ units of absorption. Let the gases proceed further through an adjoining equal length of tube $m_1\ m_2 = x_1$, then they enter at $m_1\ n_1$ with a temperature, $t_1 = t - x_1 = t\ (1 - x_1)$, above the temperature of the water; the heat taken up by the second length $m_1\ m$ will therefore be

$$H_2 = (t - x_1\, t)\, x_1 = t\, x_1 - t\, x_1^2, \text{ and}$$
$$t_2 = t_1 - H_2 = t - x_1\, t - x_1 t + t\, x^2 = t\,(1 - 2\,x_1 + x_1^2)$$
$$= t\,(1 - x)^2, \text{ the temperature at } m_2\, n_2.$$

In the same manner we find, for a third adjoining length—

$$H_3 = t\,(1 - x)^2\, x = t\,(x - 2\,x^2 + x^3); \text{ and}$$
$$t_3 = t_2 - H_3 = t\,(1 - 2\,x_1 + x^2 - x + 2\,x^2 - x^3) = t\,(1 - x)^3.$$

From these expressions the general law for the decrease of temperature throughout the length of the tube can be discerned—viz., that for a progress in length in arithmetic proportion the temperature falls in a geometric ratio.

By passing through: $x, 2\,x, 3\,x, 4\,x,$. length;
the temperature of the gases falls in proportion, as
$(1 - x), (1 - x)^2, (1 - x)^3, (1 - x)^4.$

A progression representing in arithmetic ratio numbers which are naturally in geometric ratio is well known as the table of hyperbolic logarithms. It is therefore possible to express the laws just deducted by a formula or rule embodying the relation of the temperature to the length of tube passed through as that of a number to its hyperbolic logarithm, viz.:—

$$\text{hyp. log. } t_m : \text{hyp. log. } t_n = x_m : x_n : + \ \ldots\ldots\ldots\ldots \quad (1.)$$

Let x_m = 1 from the front end of tube then is:
t_m = the temperature of the gases above that of the water, say 3,000° less the unit of absorption;
x_n = L, the whole length of tube consequently;
t_n = the temperature in the smoke box;

then, from proportion (1), the temperature of gases escaping from the end of the tube is given by the equation—

$$\text{hyp. log. } t_n = \text{L hyp. log. } t_m \ .\ .\ .\ .\ .\ .\ .\ . \quad (2.)$$

The Fig. 196 shows how the above results can be embodied in a graphic representation.

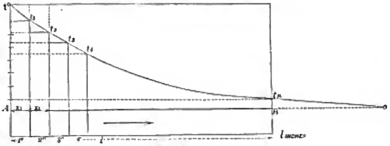

Fig. 196.

In this figure the whole tube is divided into equal lengths, x_1, x_1, say of 1 inch each. Over each of these divisions a vertical line is erected, representing by its length the temperature of the gases passing through this section. The first vertical at the front end of the tube has a length representing the initial temperature of the furnace gases entering the tube. The second vertical, being 1 inch apart, is shorter than the first, the difference being due to the heat absorbed by 1 inch length of tube at the temperature = $t°$ of the gases, or t units of absorption. This difference can be represented as a definite fraction of the length of the vertical t; for instance, $\frac{1}{8}$, or, in general, $\frac{1}{y}$.

To find the verticals for each subsequent section, we have to take the logarithm of the remaining $\frac{7}{8}$ or $\left(1 - \frac{1}{y}\right)$ and to multiply this with 2, 3, 4 in succession, this will give the logarithms of the heights for the verticals, which may be found from the logarithmic tables. In the exapple represented by the figure, where the supposed unit of absorption is $\frac{1}{8}$ of the heat contained in the gases, the logarithm of $\frac{7}{8}$ would be: log. 7 − log. 8 = − 0·134.

Therefore the logarithms of successive verticals—

$$-0{\cdot}134,\ -0{\cdot}268,\ -0{\cdot}402,\ -0{\cdot}536,$$

and the corresponding heights or temperatures—

$$\frac{1}{1{\cdot}15},\ \frac{1}{1{\cdot}30},\ \frac{1}{1{\cdot}49},\ \frac{1}{1{\cdot}70};$$

being the reduced fractions of—

$$\frac{7}{8},\ \frac{49}{64}=\left(\frac{7}{8}\right)^2,\ \frac{343}{512}=\left(\frac{7}{8}\right)^3,\ \frac{2{,}401}{4{,}096}=\left(\frac{7}{8}\right).$$

The area inclosed by the base line A B, the two verticals, and the curve $t_1\,t_n$ represents the quantity of heat absorbed or transmitted by the tube, while the narrow triangle t_n B O (which in reality is infinitely long, yet of limited area) represents the quantity of heat left in the gases and carried off by the same.

The algebraic expression for H, the total heat absorbed, is

$$\left.\begin{matrix} H_n = H_1 + H_2 + H_3 + \ldots = L\,(t_0 - t_n)\text{ in units of} \\ \text{absorption,}\end{matrix}\right\}\ \ldots(3.)$$

We have now ascertained the laws which regulate the decrease of temperature and the quantity of heat absorbed in the entire length of a *given tube*, through which a *certain quantity of gases* of a *given temperature* passes per minute; we shall next have to consider the influences upon the results which a variation of each of the above-named items can have. The laws just deducted applying in general to every tube, it is obvious that the influence of the varied conditions will be restricted to the value of *the unit of absorption—i. e.*, the heat which 1 inch length of tube will take up from gases hotter than the surrounding water by 1 degree.

Influence of the Quantity of the Furnace Gases.—Leaving the dimensions of the tube unaltered, and the initial temperature of the gases the same, an increase of the quantity of gases passing through the tube, either by augmentation of the specific weight of the gases, or by increased speed of current, will have the effect of lessening the rate of decrease in temperature. The greater quantity of gases, containing a greater quantity of heat, will require the action of a greater heat-absorbing area for the same reduction in temperature.

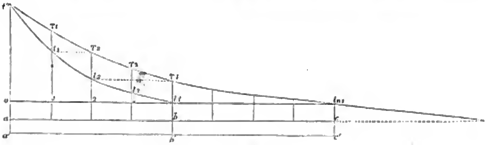

Fig. 197.

Suppose *a b*, Fig. 197, to be the original length of tube, with the quantity *g* of gases passing through it per minute, and $t_1, t_2 - t_3, t_4$, the curve representing the fall of temperature. Let the weight, *g*, of the gases be doubled = 2 *g* per minute. The result will be, that after passing through the distance of one division, and giving off the quantity of heat corresponding to this distance, there will be heat enough for absorption by a second similar length before the temperature t^1 will be reached. We must therefore double the length of run to find the temperature t_2. The curve in that case will have a slower fall, and the temperature at the end of the tube (*b*) will be as high as it was in the middle of its length before. The result, therefore, is a greater absorption of heat, as represented by the greater area *a b*, t_0, T_2, T_3, T_4; but at the same time there is a greater waste of heat, since the heat represented by the whole area T_4, C O, is thrown away. To get the same economy of fuel, we must then add the length *b c* to the tube, which is equal to *a b*. The quantity of heat absorbed by the original length of tube is increased by increasing the quantity of furnace gases; *but the increase is not in direct proportion to the quantity of gases, being reduced by the greater waste of heat carried off by the gases.*

Influence of Specific Heat of Gases.—Specific heat acts in the same manner as quantity of gases.

Influence of the Initial Temperature.—The absorption is in direct proportion to the temperature. This has been already considered, since the unit of absorption has been adopted with the supposed difference of one degree of temperature between the gases and the water. The Fig. 198 shows the influence of initial temperature upon the curve.

The dotted line represents the fall of temperature with double the initial temperature, which has a more rapid fall than the lower curve. The quantity of heat absorbed by the same tube is greater, as shown by the difference of area enclosed by the two curves, the larger being exactly double the size of the smaller. The temperature at the end (*b*) is not increased, so that the economy of fuel remains unaltered. *The heat absorbed by a tube is in direct proportion to the initial temperature of the gases, and the economy of fuel is not affected by differences of this temperature.*

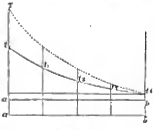

Fig. 198.

Influence of the Diameter of the Tube.—It being known that the absorption or transmission of heat is in direct proportion to the heating surface, it follows that 1 inch length of tube will absorb heat in proportion to its circumference or diameter. *The units of absorption for tubes of different diameters are therefore in direct proportion to the diameters.*

With the variation of diameter the sectional area *varies in proportion to the square of the diameter.* The diameter of the tubes therefore necessarily varies a considerable number of other conditions which have influence upon the rate of absorption of heat. With the same speed and density of gases, a tube of twice the diameter will be passed by four times the quantity of gases, which have only double the heating surface to act upon. The case is the same as the one considered before, with twice the quantity of gas passing the same tube, and causing an increased absorption with a sacrifice of economy. *To obtain the same economy from a tube of double the diameter, we must also double its length;* or, in more general terms, *the proportion between diameter and length of a tube is constant for the same economical effect in transmission of heat.* This relation has created the term calorimeter for the proportion between diameter and length of locomotive boiler tubes.

The alteration of the diameter has another still more important effect upon the transmission of heat. The gases in the interior of the tube have been hitherto considered as if each atom of the same in succession were in direct contact with the heating surface, and the temperature throughout each section the same in every place. With one and the same tube under varied conditions, or with tubes of equal diameter, this supposition is admissible, because it does not affect the ultimate results; but as soon as we compare tubes of different diameters, these conditions are altered, and it is necessary to consider the state of the gases within the tube still closer. Take any given section through the length of the tube, then the circular area occupied by the gases as the source of heat is in direct contact with the heat-absorbing surface only at its circumference; and with the small conductivity of gases, the influence of direct contact cannot extend to any considerable distance within the tube. Considering the great rapidity of the current in a locomotive boiler, there is not much probability of an interchange of position between the hotter and colder portions of the gases, which exists only in large flues, and with a slow motion of furnace gases. There remains, therefore, only one other mode of transmission of heat which is more congenial to the nature of gases—viz., *radiation.* It is well known that gases allow radiated heat to pass through very freely without absorbing any considerable quantity of the heat transmitted. The radiated heat from the sun passes the atmosphere without considerably raising its temperature until it reaches the solid body of the earth, which absorbs the heat, and slowly imparts it to the atmospheric air in contact with the earth. So a thermometer placed opposite to a flame at some distance may show a much higher temperature than another thermometer not within the influence of radiation, and surrounded by the air in close proximity to the flame.

The heat communicated by radiation decreases as the square of the distance is increased. It is also in direct proportion to the quantity and temperature of the heated gas, and to the specific heat of the same,—which three items are combined in the term *total quantity of heat* of the gas.

If we consider the whole sectional area filled with gas of equal temperature, or if we suppose it to be occupied by concentric rings of different temperatures, yet, with an equal temperature throughout each ring, the action of radiated heat will be found to follow the same general laws which have been discovered by Newton with regard to the action of gravitation, and which form the fundamental principles of all physical science. In analogy with the Newtonian laws, the action of the heat radiated from the circumference of the circular area under consideration will be represented by the total heat contained in this area collected in the centre and radiating from the same, the centre of the tube being the centre of radiation, in the same manner as it is the centre of gravity of the circular section. We may therefore consider the total heat passing a boiler tube in the furnace gases as collected in the axis, which then will represent a mathematical line endowed with the total quantity of heat which is contained in the gases, and available for evaporation.

The distance from the centre line of the tube to its circumference is equal to the radius, *and the quantity of heat which the tubular surface will receive in a given time, by radiation from the centre, will be in the inverted ratio of the square of the radius, or of the diameter of the tube.*

The Fig. 199 represents the total influence of the variation of diameter.

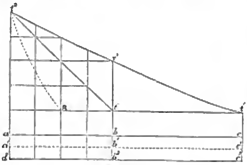

Fig. 199.

Let $a\ b$ = the original length of tube, and $t_0\ t_2$ the corresponding curve of temperature with the quantity g of gases. Double the diameter of the tube $a\ a_2$, then the quantity of gases will be $4\ g$, and the rate of absorption per minute only one-fourth of the original, or, as the heating surface is double per inch of length, the unit of absorption *will be equal to one-half the original.* The fall of temperature will be represented by the upper curve $t_0\ T_1$, and in order to obtain the same economy of heat—*i. e.*, the same temperature of escaping gases at $c\ c_2$—it will be required to double the length of the tube. In that case the tube will have four times the heating area of the original tube, and the heat absorbed per square foot of heating surface will be the same as before. For increasing the diameter of the tube it is therefore necessary to enlarge the heating surface in proportion to the square of the diameter, in order to obtain the same evaporating power and economy of fuel.

If, on the contrary, the diameter of the tube be reduced to one-half, the heat-absorbing power of the tube would be fourfold per unit of heating surface. The heating surface being reduced to one-half, the unit of absorption will be twice that of the original tube. The dotted curve, R R_1, falling more rapidly than the original, will represent the temperatures for the quantity, $= g$, of gases passing through. This curve reaches the temperature of the smoke box at half the length of the tube (a b). In order to make the second half of the tube equally efficient, it will be necessary to double the quantity of gases (g) passed through. Thus, through a tube of one-half the diameter with one-half of the heating surface, double the quantity of gases, $= 2\,g$, can be passed with the same economy and efficiency as through the original tube. Each square foot of heating surface, therefore, transmits four times the quantity of heat. *With a reduction of the diameter of the tube the evaporative effect of each square foot of heating surface increases as the square of the diameter.*

Having now determined the influence of the various elements upon the heat transmitted through a tubular heating surface, we shall be able to determine the exact value of the unit of absorption and of the total heat absorbed.

From equation (3) we have the total heat absorbed per minute—

$$H = L\,(t_1 - t_2).$$

Under the conditions: $g = 1$, a given quantity of gases.
$S = 1$ „ specific heat.
$C = 1$ „ circumference of tube section.
$d = 1$ „ diameter of tube.

Allowing all these elements to be variable, we have—

$$H = \frac{L^2}{d^2}\Big[L\,C.\,g\,S.\,(t_1 - t_2)\Big] = \Big[L\,C.\,g\,S\,t_1 - L\,C.\,g\,S\,t_2\Big]\left(\frac{L}{d}\right)^2$$

L C = the heating surface = F.

g S t_1 = the total heat contained in the furnace gases = W.

$$\text{Therefore}: H = \Big[W\,F - W\,F\,\frac{t_2}{t_1}\Big]\left(\frac{L}{d}\right)^2. \quad (4)$$

To determine the value of t_2, we have equation (2) to modify for variable g S . F . d.

$$\text{hyp. log } t_2 = L \text{ hyp. log. } t_1.$$

From the relations we found for the influence of the variables, we find the more general equation—

$$\text{hyp. log. } t_2 = \frac{F}{g\,S}\left(\frac{L}{d}\right)^2 \text{hyp. log. } t_1 \;\ldots\ldots\; (5.)$$

The two equations (4) and (5) express in mathematical form all the laws which have been found in the preceding deduction, and which may be resumed in the following:—

1. The evaporative effect of a boiler tube is proportionate to the total heat of the gases, and to the total heating surface.

2. The temperature within the tube falls in a geometric progression with an arithmetic progression of the length passed through by the gases.

3. The evaporative effect per square foot of heating surface increases as the ratio between length and diameter of the tube, or, which is the same, the ratio between heating surface and area of flue-way.

The perfect agreement of the above theoretical consideration of the locomotive boiler, with the results obtained from numerous experiments, and from the experience of long-extended practice, is a convincing proof of their correctness. The results of a considerable number of trials with locomotive engines of varied proportions and dimensions of boilers, made by different authorities have been compiled by Mr. Clark. An evaporative duty of about 9 lbs. of water for 1 lb. of coke has been accepted as a standard of economy, and all experiments showing that degree of useful effect have been compared with each other. The rate of consumption of fuel has been estimated per square foot of grate area, which area generally is proportionate to the flue-way or total area of tubes. For facility of comparison, the heating surface is also expressed as a multiple of the grate area, the quotient obtained by dividing the two areas being called *their ratio.*

Observations of this kind have been made by Stephenson, Vignoles, Pambour, D. Gooch, D. K. Clark, Sinclair, and others. From all these the best results have been selected, as furnishing proportions suitable for imitation; and the generalization of these proportions, expressed as a rule or formula, has been afterwards relied upon as a standard for good practice.

The following table has been compiled by Mr. D. K. Clark, as comprising the most favourable results obtained with locomotive boilers:—

RELATIVE HEATING SURFACES AND RATES OF CONSUMPTION.

Ratio of Heating Surface to Grate Area.	Consumption of Water per Hour per Square Foot of		Water Evaporated per Pound of Coke.
	Grate.	Heating Surface.	
	Cubic Feet.	Cubic Feet.	Lbs.
52	6·15	·12	9·
66	8·	·13	9·1
72	12	·17	8·9
90	18	·20	8·92

The results given in the above table admit of graphic representation.

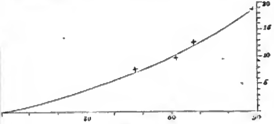

Fig. 200.

The base line in Fig. 200, measures the ratios of heating surface as taken from column 1. The vertical heights of the stars from the base line represent the consumptions as noted in the second column. They also represent the consumption of coke per square foot of grate area, since that is proportionate to the evaporation of water, as shown by the approximate equality of the figures in the last column.

The curve is the graphic expression of the law which regulates the proportion between ratio of heating surface and consumption of fuel, or evaporation of water. Upon close inspection of this curve it will be found that the *increase of consumption is equal to the square of the increase of ratio.*

These relations, deduced entirely from practice and experiment, are precisely the same which we have found by the foregoing theoretical researches. The element of grate area represents in this instance that of the flue-way, which, in all locomotive boilers subjected to this investigation, bears the strict proportion of about ¼ to 1 to the grate area. It is obvious, therefore, that the ratio between heating surface and grate area is proportionate, under the existing circumstances, to the ratio of heating surface and flue-way. Whether or not the relation of grate area and tube area should be strictly constant is a subject for future consideration; it will therefore be more advisable to make the formulas or rules for proportioning locomotive boilers independent of the grate area, and confine them to those elements which are proved directly influential by theory. For this purpose we shall have to substitute the area of flue-way—*i. e.*, the collective sectional area of all tubes—for its equivalent value of grate area (g), viz.:—

for g we substitute $= 4\,w$.

$w =$ area of tubes. Then, instead of the ratio r, we shall have the ratio $\frac{h}{w}, = C$. The fire box not being under consideration at present, this ratio is identical with the fraction—

$$\frac{h}{w} = \frac{n\,.\,d\,L\,.\,3{\cdot}141}{n\,.\,d^2\,.\,3{\cdot}141} = \frac{L}{d} = \frac{\text{length of tubes.}}{\text{diameter of tubes.}}$$

The fraction $\frac{L}{d}$, being the ratio between the length of each boiler tube and its diameter, has been termed the *Calorimeter;* and the general law, confirmed by theory and practice, may be expressed as follows:—

The evaporative power of the total heating surface is proportionate to the square of the calorimeter, and the evaporative power of each square foot of heating surface is proportionate to the calorimeter.

Conclusions for proportioning the different elements of a locomotive boiler.

The problem of constructing a locomotive boiler in its most general form contains certain given conditions and restrictions dictated by practice. These are, in the first instance, the limits for length and width of the whole structure; of which the first is regulated by the extent of admissible wheel-base, the second by the gauge of the railway. The next limit imposed upon the construction is the total weight of the boiler, which must be within the load that can be carried upon the wheels without injuring the rails and tyres. With the outer dimensions limited as they are, there is, however, not much danger of exceeding the limits for weight, so long as the construction resembles the standard types of present practice. A further condition given at the outset is the nature and quality of the fuel, and with this the necessary supply of air; also the relative quantities to be introduced through and above the grate for each pound of fuel burnt are fixed. The demand for steam and the economy of fuel may be considered unlimited in one sense, and only the minimum of each is pointed out by the most successful examples of previous practice. At present the rate of 9 lbs. of cold water evaporated per pound of best coke, and the consumption of about 13·5 lbs. of water per square foot of total heating surface, are considered the maximum recorded in practice. It must be borne in mind that the two items—economy of fuel and efficiency of heating surface—are opposed to each other; that an increase in the one beyond the exact quantity due to the proportions of the boiler is followed by a decrease of the other; and that, therefore, each boiler has a certain maximum efficiency, where the highest rate of evaporation coincides with the greatest practicable economy of fuel. This maximum rises as the square of the calorimeter; and it will therefore be the first aim in constructing a locomotive boiler to obtain the greatest possible length and the smallest diameter of tubes admissible. In this respect the limits dictated by practice are somewhat embarrassing. The length of tubes is kept closely limited by the total length of boiler fixed beforehand; there remains only the reduction of the diameter of tubes.

Here again the limits are very narrow. As we reduce the diameter of tubes, of course their number increases, since we shall always try to get the utmost extent of heating surface compatible with efficiency. In order to have the same extent of heating surface with tubes of half the diameter, we must double their number. Still their aggregate area or flue-way will be only one-half, and the current of gases will have to pass it with double the speed. But this is not all. As we double the calorimeter, the economical effect of evaporation per square foot of heating surface is doubled. The same heating surface will produce twice the quantity of steam, and therefore require twice the absolute quantity of furnace gases per minute. The speed of the current, doubled already by the narrow flue-way, must be still further doubled, giving a total of four times the speed due to the original calorimeter. The current of furnace gases or draught in the locomotive boiler is produced by the blast, and, as will be shown more explicitly hereafter, requires a higher blast pressure for its greater speed; and this can be had only at the expense of useful steam pressure. It is likewise to be borne in mind that, with a constant proportion of grate area and flue-way, the grate is reduced to one-half its original size by doubling the calorimeter. The quantity of fuel to be burnt on that reduced area being multiplied by two, it follows that on the square foot of grate area four times the quantity of fuel is to be burnt per minute. This still further imposes strict limits to the reduction of flue-way, since it is practically impossible to exceed a certain rate of combustion per square foot of grate. The necessity of frequent firing, the depth of incandescent fuel upon the grate, and the resistance offered to the air passing through it, are the practical obstacles to increased efficiency in that direction.

Mr. Clark has found that in well-managed boilers good coke may be properly consumed at a rate of 100 to 112 lbs. at least per hour per foot of grate, without involving any objectionable consequences to the boiler. As a general rule for practice, he recommends a consumption of 112 lbs. of coke per hour per foot of grate to be the maximum at which locomotive boilers should be designed to work.[*]

The question would now arise, whether the ratio between grate area and flue-way is necessarily fixed. With boilers using coke as fuel, the quantity of gases produced by 1 cubic foot of coke is fixed; therefore, with a certain speed of current and a given thickness of fuel upon the grate, the area of the latter is strictly fixed. Still, the supposition of a fixed thickness of fuel upon the grate does not exist in practice. The thickness of layer varies between a maximum existing after a fresh charge of fuel has been thrown upon the grate, and a minimum immediately before addition of a fresh charge. For uniformity of combustion and evaporation, it is desirable that the fluctuations between this maximum and minimum should be as small as possible; and in order that the open fire door may not admit an undesirable quantity of cold air into the fire box, the necessity for charging should not arise too often. All this shows that a large grate area is desirable to co-exist with a small flue-way. On the other hand, we have the grate area limited in width by the gauge, and limited in length by the position of the wheels and their axles, and still more limited by the consideration that the addition to the length of grate and fire box shortens the tubes and diminishes the absolute extent of heating surface.

The following table shows how these elements have influenced each other in the engineering practice of the last twenty years:—

TABLE SHOWING THE GRATE AREA, FLUE-WAY, AND CALORIMETER OF LOCOMOTIVE BOILERS BURNING COKE.

Date.	Railway or Maker.	Grate Area.	Flue-Way.	Flue-Way per Unit of Grate.	Calorimeter.
		Sq. Ft.	Sq. Ft.		
1840	Stephenson, . . .	9·46	1·46	0·15	56·3
1845	E. & G. R., . . .	10·92	2·06	0·19	70·
1850	M. S. & L. R., . .	10·56	2·96	0·28	80·
1862	G. England Co., . .	8·5	3·01	0·35	58·
"	Borsig, ...	11·6	3·01	0·26	85·

From this table it appears that the calorimeter has been increased with the progress of locomotive engineering, and that the grate area has also changed its ratio to the flue-way in the tubes, but not in a similar proportion. With locomotive boilers designed for combustion of coal these proportions are different: 1 cubic foot of coal produces about 1¾ times the quantity of furnace gases compared with 1 cubic foot of coke; the same absolute quantity of fuel would therefore occupy a smaller grate, with an equal depth of layer upon the grate. This, however, is not the course followed in practice, since the advantageous combustion of coal requires a thin layer of fuel upon the grate, and a slow motion of air through the incandescent fuel, and through the fire box. This is necessary in order to give the coal gas, or hydrocarbon, time to be mixed with the oxygen of the air, and to effect its complete combustion before the gases enter the tubes. Combustion cannot be carried on within the tubes to any considerable extent. The cooling influence of the tube surface is so great as to extinguish the flame after a short distance passed through, in the same manner as the wire gauze of a miner's safety lamp extinguishes the flame of the gases within, and prevents ignition of the otherwise combustible gases. If the hydrocarbon gas produced by the coal when thrown upon the grate is carried into the tubes before it has time to effect its complete combustion, it will be carried into the smoke box unconsumed, and will be wasted. All this necessitates a comparatively slow motion of gases within the fire box, and a considerably greater extent of the fire box itself. Both advantages are obtained by enlarging the grate area. The gaseous products of combustion are of lower temperature, and consequently of less bulk, than those from coke; their flue-way may therefore be made narrower in proportion: the specific heat of the gases from coal is greater than that of coke gases; therefore, with the same speed of current, their useful effect will extend through a greater length of tube. Both these qualities allow a greater calorimeter to be adopted. With a larger grate and narrower flue-way the proportions of coal-burning locomotives are considerably different from those designed for coke. The following table shows the dimensions of some modern coal-burning locomotive boilers:—

TABLE SHOWING THE GRATE AREA, FLUE-WAY, AND CALORIMETER OF MODERN LOCOMOTIVE BOILERS BURNING COAL.

Date.	Railway or Maker.	Grate Area.	Flue-Way.	Flue-Way per Unit of Grate.	Calorimeter.
		Sq. Ft.	Sq. Ft.		
1862	London N.-W. R., .	25·	4·1	0·16	59
"	Do. Do.	14·9	3·8	0·25	69
"	Beyer, Peacock, & Co.,	18·	4·7	0·36	66
"	Sharp, Stewart, & Co.,	27·5	4·2	0·15	65
"	North of France R., .	28·2	4·9	0·17	88

Rules for Proportioning a Locomotive Boiler.

Let f = the number of cubic feet of cold water to be evaporated per square foot of heating surface, C = the calorimeter.

The fuel to be coke.

Take the calorimeter = 80, with an evaporation of 0·2 cubic feet of water per hour per square foot of heating surface as the standard, being the highest result recorded in practice, with a duty of 9 lbs. of water per pound of coke;[*] then, as the evaporation varies with the calorimeter,

$$80 : C = 0{\cdot}2 : f \text{ or}$$

$$f = \frac{0{\cdot}2}{80} \cdot C = 0{\cdot}0025\ C.$$

The calorimeter C being equal to the length of tubes (L) divided by their diameter (d).

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[*] *Railway Machinery*, p. 163.

[*] D. K. Clark, *Railway Machinery*.

$\mathrm{C} = \frac{\mathrm{L}}{d}\cdot$ by substitution in the above.

$$f = 0{\cdot}0025\,\frac{\mathrm{L}}{d} \ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(1.)}$$

Let F = the total quantity of water evaporated per hour in cubic feet by the whole heating surface, h; then

$$\mathrm{F} = h \cdot f \cdot \text{ or from (1)}$$

$$\mathrm{F} = h \times 0{\cdot}0025\,\frac{\mathrm{L}}{d} = 0{\cdot}0025\,h \cdot \frac{\mathrm{L}}{d} \ldots\ldots \text{(2.)}$$

$$h = \frac{1}{0{\cdot}0025}\,\frac{d}{\mathrm{L}} \cdot \mathrm{F} \cdot = 400\,\mathrm{F} \cdot \frac{d}{\mathrm{L}} \ldots\ldots\ldots \text{(3.)}$$

Let $r = \frac{h}{g}$ = the ratio of heating surface and grate area; f_1 = the cubic feet of water evaporated per square foot of grate; then

$$f_1 = \frac{\mathrm{F}}{g}$$

and $g = \frac{h}{r}$; therefore by substitution

$$f_1 = \frac{\mathrm{F}\,r}{h} \cdot \text{ by substitution of F from (2.)}$$

$$f_1 = 0{\cdot}0025\,\frac{h \cdot \mathrm{L} \cdot \mathrm{F}}{d \cdot h} = 0{\cdot}0025\,r\,.\,\frac{\mathrm{L}}{d} \ldots\ldots \text{(4.)}$$

With the supposition that in modern practice the area of grate bears the ratio to the flue-way of tubes as 4 to 1, which is the proportion to be recommended for imitation in the generality of cases, and only to be deviated from if some special object is to be attained, we get a series of simpler approximate formulas more specially adapted for designing locomotive boilers in strict accordance with our standard types.

For that special type the ratio r is nearly the same as the calorimeter of the tubes, and therefore r may be substituted for $\frac{\mathrm{L}}{d}$

Let again f_1 = the cubic feet of water evaporated per hour per square foot of grate.

Take the ratio r from the same boiler which has been proposed as standard, $r = 90$, then we may with sufficient accuracy consider the cubic feet to vary with the square of the ratio, since from (4.) we find

$$f_1 = 0{\cdot}0025\,r^2;$$

and from the proportion

$$90^2 : r^2 = 18 : f_1, \text{ we have}$$

$$f_1 = 0{\cdot}00222\,r^2,$$

the slight discrepancy being due to the area of fire box, which in the second case is included in the ratio, while in the more general formula it does not unduly influence the results.

The ratio r being equal to $\frac{h}{g}$, we find by substitution

$$f_1 = 0{\cdot}00222\left(\frac{h}{g}\right)^2. \ldots\ldots\ldots\ldots\ldots \text{(1.)}$$

Let f = the evaporation per hour per square foot of heating surface;

$$f = \frac{f_1}{r} \text{ or } f_1\,r = f, \text{ substituting in equation (1.)}$$

$$f\,r = 0{\cdot}00222\left(\frac{h}{g}\right)^2, \text{ and since } \frac{h}{g} = r$$

$$f = 0{\cdot}00222\,\frac{h}{g}. \ldots\ldots\ldots\ldots\ldots\ldots \text{(2.)}$$

Let F, the total consumption of water per hour, then

$$\mathrm{F} = f_1\,g \text{ or } \mathrm{F} = f\,h.$$

By the substitution of either of these expressions in the equations (1.) and (2.) respectively, we have

$$\mathrm{F} = 0{\cdot}00222\,\frac{h^2}{g} \ldots\ldots\ldots\ldots\ldots\ldots \text{(3.)}$$

From formula (3.) we find

$$h^2 = \frac{\mathrm{F}\,g}{0{\cdot}00222} = 450\,\mathrm{F}\,g;$$

$$\text{from which } h = \sqrt{450\,c\,g} = 21{\cdot}2\,\sqrt{\mathrm{F}\,g} \ldots \text{(4.)}$$

From formula (3.) we can also deduct

$$g = 0{\cdot}00222\,\frac{h^2}{\mathrm{F}}. \ldots\ldots\ldots\ldots\ldots\ldots \text{(5.)}$$

The above five formulas are the mathematical expressions for the following rules:—

Rules 1, 2, 3.—*To find the greatest rate of consumption of water consistent with its economical evaporation for a given heating surface and grate area.*

1. *The rate of consumption per square foot of grate area.*—Divide the heating surface by the grate area, both in superficial feet, square the quotient, and multiply by 0·00222. The product is the consumption in cubic feet per hour per square foot of grate.

2. *The rate of consumption per square foot of heating surface.*—Divide the heating surface by the grate area, both in square feet, and multiply by 0·00222. The product is the consumption in cubic feet per hour per square foot of heating surface.

3. *The Rate of total Consumption.*—Divide the square of the heating surface by the grate area, both in square feet, and multiply by 0·00222. The product is the total consumption in cubic feet per hour.

Rule 4.—*To find the heating surface necessary to maintain a given hourly consumption of water economically with a given area of grate.*

Multiply the grate area in square feet by the consumption of water in cubic feet per hour; find the square root of the product, and multiply the root so found by 21·2. The final product is the area of heating surface.

Rule 5.—*To find the grate area suitable for maintaining a given hourly consumption of water economically with a given heating surface.* Divide the square of the heating surface in square feet by the consumption of water in cubic feet per hour, and multiply by 0·00222. The product is the area of grate in square feet.

For coal-burning locomotives rules of this kind would be dangerous to use, since no ratio of grate and flue-way can be fixed, owing to the great difference in the chemical composition of various kinds of bituminous coal, and to the different modes for introducing the air into the fire box. It will be advisable to increase the total heating surface of coal-burning locomotive boilers by the same per centage, as the temperature of the products of combustion, calculated from an analysis of the coal, is lower than the temperature of the coke gases.

Clearance.—There is one most important item in the construction of a boiler yet to be considered—viz., the clearance between the tubes and between tubes and boiler shell. The clearance is wanted to allow a sufficiently quick circulation of water and steam around and between them, so as to let the steam escape as quickly as it is formed, and prevent its remaining in contact with the tube, and in raising its evaporative effect. Researches on the physical laws upon which the circulation in a locomotive boiler depends have not been published as yet; and in the total absence of any scientific investigation, we have nothing to rely upon as a guide in practice but a few empirical rules given by the best authorities in locomotive engineering.

Mr. Clark's rule is the following:—To find the clearance between the tubes suitable for economical evaporation for a given number of tubes. Divide the number of tubes by 30. The quotient is the required clearance in eighths of an inch.

Accordingly the clearance ought to be for

Number of Tubes	Clearance.	Number of Tubes.	Clearance.
Below 90 —	3/8 inch.	150 — 165	11/16 inch.
90 — 105 —	7/16 "	165 — 180	3/4 "
105 — 120 —	1/2 "	180 — 195	13/16 "
120 — 135 —	9/16 "	195 — 210	7/8 "
135 — 150 —	5/8 "	210 — 225	15/16 "

The lateral clearance between the upper rows and the barrel should be about 1/15 of the diameter of the barrel.

Some French authorities allow less clearance. The Society of Engineers in Berlin recommend ⅛th of an inch for every 50 tubes.

Influence of the Rate of Combustion.—We may now proceed to examine the total heat transmitted through the entire heating surface of a locomotive boiler, and the proportionate share in the evaporation which is due to its different component parts in a special example, with varied rates of combustion.

Call B the weight of fuel burnt per hour upon the grate.

H, the total heat produced per lb. of fuel by complete combustion.

W_1, the heat transmitted through the fire box.

W_2, the heat transmitted through the tubes.

$W = W_1 + W_2$, the total heat transmitted to the water in the boiler.

F_1, the area of fire box.

F_2, the total heating surface of the tubes.

F, total heating surface.

$$\text{Let } F_1 = 20 \text{ square feet, } F_2 = 240,$$
$$F = F_1 + F_2 = 260 \text{ square feet.}$$

Quantities of Heat.	Consumption of Fuel in Lbs. per Hour.				
	144	216	324	468	648
$W_1 : B\,H$	0·1829	0·1264	0·0862	0·0607	0·0442
$W_2 : B\,H$	0·6811	0·6567	0·5768	0·4785	0·3902
$W : B\,H$	0·8640	0·7821	0·6630	0·5392	0·4344
$W_2 : W_1$	3·7	5·19	6·68	7·88	8·60
$\frac{W_1}{F_1} : \frac{W_2}{F_2}$	3·22	2·31	1·79	1·53	1·38
Pounds of water evaporated by 1 lb. of coke,...	10·9	9·93	8·4	6·8	5·4

The preceding table, compiled by Redtenbacher,* will show the relative performances of each part of the boiler.

From this table it will be seen that the useful effect of the boiler, or the per centage of the heat utilized by it, decreases with an increasing rate of combustion. With a consumption of 144 lbs. of fuel per hour the boiler utilizes 86 per cent. of the heat generated, since

$$W = 0{\cdot}8640\ B\ H.$$

With an increased consumption, say 648 lbs. per hour, the total heat transmitted to the water in the boiler is only 43 per cent. of the heat produced by combustion.

The highest rate of evaporation per lb. of coke, as shown in the table, is somewhat higher than that which is found by experiments; this is due to the neglect of the effects of priming, and other drawbacks, which may amount to 10 per cent. of the figures given.

The fifth horizontal line in the table, showing the relation of $\frac{W_1}{F_1} : \frac{W_2}{F_2}$, gives the proportion of the evaporative power of a square foot of the fire-box surface, compared with a square foot of tube surface. This relation is not constant, but depends upon the rate of consumption of fuel. With slow firing the fire box is more effective in proportion; but with an increased quantity of heat generated in a given time, a larger proportion is left for absorption by the tubes.

To compare the evaporative duties of the different lengths of each tube with each other, take for example the consumption of coke per hour, 324 lbs., with the above proportions of fire box and tube surface.

The proportionate quantities of heat utilized by

$$\begin{array}{lcccc} & 60, & 120, & 180, & 240 \text{ square feet} \\ \text{are } \frac{F_2}{B\,H} = & 0{\cdot}2118, & 0{\cdot}3695, & 0{\cdot}4869, & 0{\cdot}5742. \end{array}$$

These figures represent the proportions of heat transmitted through ¼, ½, ¾, 1 tube surface.

By subtraction we find the proportionate evaporative duty.

For the first quarter of the length of tubes, . .	0·2118
" second " " " " . .	0·1577
" third " " " " . .	0·1174
" last " " " " . .	0·0873

With this rate of combustion the evaporation due to the first quarter of the length of tubes is 21 per cent.; that of the second quarter, 16 per cent.; of the third, 12 per cent.; and of the last quarter, 8 per cent. of the total evaporation. The small per centage of duty obtained from the last section shows that there is no further practical economy to be expected from additional heating surface, unless the rate of combustion be increased.

With a combustion of 648 lbs. of coke, on the contrary, the four sections of the length of tubes in the above described boiler would evaporate nearly equal quantities of steam, which shows that an addition of heating surface would effect a further utilization of heat with that increased rate of combustion. The above example will suffice for demonstrating the intimate connection which exists between the proportions and sizes of the different elements of a locomotive boiler and the rate of combustion, and to show that the limits of evaporative power of the boiler cannot be exceeded in practice except at the expense of economical effect or evaporative duty.

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* The formulæ from which this table is calculated require for their deduction the assistance of the differential and integral calculus. Those readers who are interested in the subject, we refer to Professor Redtenbacher's celebrated work, *Gesetze des Locomotiv Baues*. Mannheim, 1855.

CHAPTER XV.*

THEORY OF THE BLAST.

The supply of air required for the combustion in a locomotive fire box is very considerable. In modern locomotives from 1,200 to 2,000 cubic feet of cold air pass through the fire box per minute. This great volume is expanded to about ten times its bulk at the moment it passes the fire box ferules, or the front entrance of the tubes, and it occupies about double its original volume when it leaves the smoke box, escaping through the chimney with the exhaust steam. The resistances to the motion of so great a volume of air through the restricted passages offered by the grate, the tube area, and the chimney, must of necessity be considerable; and it requires a very powerful agent to create an artificial draught sufficiently strong to overcome this resistance.

It is well known that the ingenious and simple expedient of creating the artificial draught in a locomotive engine is the passing of the exhaust steam from the cylinders into the chimney through a pipe having its orifice placed concentrically with the former. This pipe is termed the blast pipe, and it forms one of the most prominent features of the locomotive engine, and at the same time one of the most perfect of its component parts. It has been left almost unaltered ever since its first application; and its action, when properly proportioned to the other working parts of the engine and boiler, leaves so little to be desired that there is very little probability that future progress of locomotive engineering will make vital changes in this simple and beautiful apparatus, unless, indeed, further use should be made of the exhaust steam, and the principle of exhausting into the atmosphere be discarded at some future time.

It is, however, much easier to admire the action of the blast pipe than to understand its mechanical principles; and in this, as in many other instances, practice has led the way almost unassisted by theory, while the latter has only at a very recent date arrived at a satisfactory explanation of what had been done for forty years past in the universal routine of locomotive construction. The evidence of direct observation shows that the passage of the exhaust steam up the chimney creates a partial vacuum in the smoke box. A vacuum gauge fixed to the smoke box, and connected with its interior, permits of a distinct measurement of this vacuum under varied circumstances and conditions.

It is likewise an object of direct observation that the jet of steam escaping from the orifice of the blast pipe diverges conically until it reaches the sides of the chimney, which it fills throughout its entire section.

The vacuum created in the smoke box is the effect produced by the passage of steam from the narrow blast orifice into the wider sectional area of the chimney. The action of the blast will therefore be governed by the same general laws which regulate the flow of liquids through passages of varying sections—laws which have been very closely investigated at a recent date by Combes, Rankine, Weissbach, Turner, and other authorities.

This has been principally due to the invention of the Giffard Injector, which has drawn the attention of the scientific world to this interesting but somewhat difficult subject. The theory of the blast pipe, and that of the injector, are very closely related, both appliances being in fact applications of the same mechanical principles under varied conditions.

The Flow of a Liquid through Passages of Varying Sections.—Let fig. 200 represent the longitudinal section of a passage for water, air, or steam, which may be supposed to enter the cylindrical pipe A with a given velocity, *n*. At *a a* the section of the pipe is suddenly enlarged, or, in other words, the liquid passes into a pipe of larger diameter, B, from which it escapes into an absolute vacuum. It is obvious that the velocity of motion in the wider pipe, B, will be different from that in the pipe A, since the quantity which passes through the narrower section in a given time, and that which passes through the wider area, must be the same; or if the velocity remain the same in B as in A, there must be an expansion of the liquid, which will enable it to occupy a volume proportionate to the difference of sections passed through.

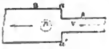

Fig. 200.

The first will be the case with water, which is not capable of expansion, and which will therefore sustain a loss of velocity in the wider passage. The second will take place to a certain extent with gases, which will alter their pressure and density on entering the wider pipe, B.

The liquid column in motion through the pipe A represents a certain quantity of work which is equal to the weight of the column multiplied by the head of water corresponding to its velocity. Suppose that one pound of water passes through A per second with a speed of 80 feet. The quantity of work contained in that liquid column will be equal to the weight of one pound raised to the height which corresponds to the above velocity, and which is $\frac{6400}{64}$, or equal to 100 feet.

Suppose the pipe B to have four times the sectional area of the pipe A, then the quantity of one pound of water which will pass through it per minute will have a speed equal to one quarter of 80, or 20 feet per second, corresponding to a head of water of only 6·25 feet. The labour performed for passing the mass of water through the wider pipe is therefore only 6¼ foot pounds, and a difference of 93¾ foot pounds, remaining from 100 foot pounds which existed in the narrow pipe, is set free by the change of velocities. These 93¾ foot pounds represent a surplus labour available for giving motion to other liquid matter, if within reach. Supposing that, at the commencement of the operation, the pipe B had been filled with water or air, this surplus of labour, or a part of it, would have been spent in giving motion to that liquid mass, and the result would have been a partial vacuum in the tube B. If now an opening be made in the side of B, as shown by the dotted circle *n*, Fig. 200, and this opening connected with a reservoir containing water, or placed in direct contact with the atmosphere, the supply of liquid for taking up the surplus work will be continuous, and so will be the action of the liquid column. The result will be a certain speed of motion in the pipe B, through which the total liquid column from the pipe A will pass, together with a proportionate quantity of liquid drawn into the tube through the opening *n*, and a partial vacuum in B, the amount of which will depend upon the size of the opening, *n*, and the resistance offered to the entrance of the liquid mass into the pipe B.

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* By Mr. Ferdinand Kohn.

Suppose now, instead of water being the element in motion, the pipe A to be a passage for steam under a given pressure and of corresponding density, then upon entering the pipe B, of four times the sectional area, the steam will have the possibility to expand. Now, in the same proportion as expansion takes place, the density of the steam is diminished, and accordingly the work required for its motion at the given velocity will be diminished. The surplus work contained in the liquid column in motion creates the tendency to impart motion to other quantities of liquid at rest, and in communication with the enlarged passage.

The liquid drawn into the current by the suction of a steam jet may be air, as in the chimney, or it may be water, as in the injector: both cause and result will be the same in either case, modified only by the nature and specific qualities of the substances operated upon.

The above consideration, although inadequate for a complete explanation of the process taking place in practice, is the simplest form in which we have been capable of expressing the fundamental principle of the mechanical action of the steam jet. It is far too primitive to serve as a basis for calculations; but it may be of some utility in giving to the reader a general impression of the action of the blast, and of the reason why a jet of steam imparts motion to the surrounding fluid.

The Mechanical Action of a Steam Jet.—Let Fig. 201 represent a vessel filled with air, and in communication with the atmosphere through the pipe D. The pipe A to be connected with the steam space of a boiler, and steam passing through it under the pressure *p*, and with the velocity *w*, corresponding to the difference of pressure in the boiler and in the vessel C. The pipe B to lead into the atmosphere, and to be of larger diameter than the orifice of the steam pipe.

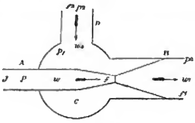

Fig. 201.

The speed, *w*, with which steam of the pressure *p* will rush into a space having the pressure p_1 is found by the formula—

$$w = \sqrt{2\,g\,\frac{p - p_1}{\delta}} = 8{\cdot}01\,\sqrt{\frac{p - p_1}{\delta}}.\text{*}$$

δ representing the specific weight of the steam.

$$\text{This gives } \frac{w^2}{2\,g} = \frac{p - p_1}{\delta} = \text{L}$$

which represents the labour, L, contained in one pound of steam moving with the velocity *w*.

On entering the space C the steam alters its speed, its pressure, and its density. The speed w_1, the pressure $= p_1$, and the specific weight $= \delta_1$.

The sudden alteration of the speed of a liquid column in motion is attended by a loss of power which has been approximatively ascertained by Carnot, and expressed by the formula—

$$\text{Loss of labour : } \text{L}_1 = \frac{(w - w_1)^2}{2\,g}.$$

This amount of labour, which disappears with every sudden change of velocity in a somewhat similar manner as a certain quantity of *vis viva* is lost by the concussion of inelastic bodies, seems to be converted into heat.

There appears still another quantity of labour in the steam moving through pipe B. The pressure p_1, within the vessel C is kept constant during the operation. A liquid moving under constant pressure receives a certain quantity of work during its motion. Suppose the cylindrical vessel A, Fig. 202, to contain a liquid mass which escapes through the opening *m* at a velocity w_1, under a constant pressure equal to p_1. We can imagine this constant pressure to be exerted by a piston forcing the liquid out of the vessel, and following it as it escapes. The space travelled through by the piston will be equal to the volume of liquid discharged, and the labour performed by the piston, which is absorbed or taken up by the liquid in motion, is equal to the area of piston, multiplied by the pressure, and multiplied by the stroke or length of space travelled through. Now, the product of the area of piston and the length of its stroke is equal to the volume of liquid expelled; and suppose this to be the volume occupied by 1 pound of the liquid, and equal to *v*, the labour exerted by the piston, and transmitted to 1 pound of liquid in motion, will be $= p_1\,v$. Against this amount of labour gained, the effect of the constant resistance of the atmosphere at *m* is to be balanced. Calling the atmospheric pressure p_0, the quantity of work performed or given out by 1 pound of liquid in overcoming it, will be $= p_0\,v$. The quantity of labour remaining to the liquid will therefore be

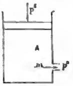

Fig. 202.

$$\text{L}_2 = p_1\,v - p_0\,v = v\,(p_1 - p_0).$$

v, being the volume of 1 pound of liquid, may be expressed by the specific weight of the same $= \delta_1$—

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* See p. 110.

$$v = \frac{1}{\delta_1}; \text{ therefore,}$$

$$L_1 = \frac{p_1 - p_0}{\delta_1};$$

the labour due to the constant pressure, p_1, within the vessel C. This expression for L_1 is only approximatively correct, since it is based on the condition that there should be no change of density in the passage of the steam from the pipe A into the vessel C, while in practice such a change from the density δ to the density δ_1 does take place. Within the limits existing in practice, however, the error is inconsiderable, and may be neglected.

The quantity of labour contained in 1 pound of steam moving from the blast orifice A, through the pipe B, is therefore the sum of the three expressions, $L + L_1 + L_2$; and if we call M the number of pounds of steam passing through the apparatus in a given time, the total work contained therein is

$$W = M\left\{\frac{w^2}{2\,g} - \frac{(w - w_1)^2}{2\,g} + \frac{p_1 - p_0}{\delta_1}\right\}.$$

The steam in passing through C, and changing its velocity, will impart motion to a certain quantity of air, which will enter the vessel C through the pipe D. Call M_1 the quantity (by weight) of air drawn in by the jet; then this weight of air when in motion will contain the quantity of work—

$$W_1 = M_1\left\{\frac{w_0^2}{2\,g} - \frac{(w_0 - w_1)^2}{2g} + \frac{p_1 - p_0}{\delta_1}\right\};$$

w_0 being the speed with which the air may be moving towards the opening D, which, in the generality of cases, is $= 0$; δ_1 is taken as the density of the mixture of air and steam in the vessel C, under the pressure p_1.

Under the supposition of the air being at rest at the mouth of pipe D, or—

$$w_0 = 0, \text{ we have}$$

$$W_1 = M_1\left\{\frac{p_1 - p_0}{\delta_1} - \frac{w_1^2}{2\,g}\right\}.$$

The total work contained in both air and steam is

$$W + W_1,$$

and that must be equal to the expression,

$$\left(M + M_1\right)\frac{w_1^2}{2\,g},$$

which likewise represents the total labour contained in the moving mass. We have therefore,

$$\left(M + M_1\right)\frac{w_1^2}{2\,g} = \left\{\frac{w^2}{2\,g} - \frac{(w - w_1)^2}{2\,g} + \frac{p_1 - p_0}{\delta_1}\right\} M$$
$$+ \left\{\frac{p_1 - p_0}{\delta_1} - \frac{w_1^2}{2\,g}\right\} M_1.$$

From this equation the following expression is found by reduction :—

$$p_0 - p_1 = 2\,\delta_1\left\{\frac{M}{M + M_1} \cdot \frac{w\,w_1}{2\,g} - \frac{w_1^2}{2\,g}\right\},$$

as the formula for the vacuum produced by the steam jet, since $p_0 - p_1$ is the difference between the atmospheric pressure outside and the pressure, p_1, inside the vessel C.

The relative quantities of steam and air, M and M_1, depend upon the proportionate size of the blast orifice, the suction pipe D, and the escape passage B.

Call the area of blast orifice $= F$;
Call the area of air passage $= F_2$;
Call the area of escape passage (chimney) $= F_1$;

then the quantity of steam passing through the blast orifice per second, and acquiring the density δ_1 within the chamber C, is—

$$M = F\,w\,\delta_1,$$

and the quantity of steam and air mixed which passes through the pipe B is—

$$M + M_1 = F_1\,w_1\,\delta_1; \text{ therefore,}$$

$$\frac{M}{M + M_1} = \frac{F\,w}{F_1\,w_1}.$$

Call the proportion of chimney and blast orifice

$$\frac{F_1}{F} = m,$$

and the proportion of air passage and blast orifice

$$\frac{F_2}{F} = n,$$

then the expression $\frac{M}{M + M_1}$ is equal to $\frac{w\,\delta}{m\,w_1\,\delta_1}$, and by substitution in the former value for the vacuum, we find

$$p_0 - p_1 = 2\,\delta_1\left\{\frac{w^2}{2\,g\,.\,m} - \frac{w_1^2}{2\,g}\right\}. \text{ ....(1.)}$$

$$\text{Vacuum} = \frac{\delta_1}{g}\left(\frac{w^2}{m} - w_1^2\right) = \frac{1}{32}\,\delta_1\left(\frac{w^2}{m} - w_1^2\right).$$

This expression shows that the vacuum will increase with the speed and density of the steam, and it will be diminished by a proportionately wide escape pipe or chimney.

Beyond this general relation, however, there can be little use made of it in the above form, since it expresses the vacuum as a function of the velocity of motion through the steam pipe and escape pipe. These velocities are themselves dependent upon the amount of vacuum created in the vessel C, Fig. 201; and since, with a given proportion of the blast orifice, air passage, and chimney, and a given steam pressure, the vacuum is actually fixed in practice, the expression for calculating the vacuum should not contain any other items, but permit of the vacuum to be determined direct from those elements.

For arriving at this end we call w_2 the velocity with which the air is moving through the pipe D, then the quantity of air drawn into the chamber C,

$$M_1 = F_2\,w_2\,\delta_1.$$

The contraction and other resistances being neglected, the same as throughout the whole present deduction,

It follows from,

$$M + M_1 = F_1\,w_1\,\delta_1 \text{ and } M = F\,w\,\delta_1,$$

that

$$F_1\,w_1\,\delta_1 = F\,w\,\delta_1 + F_2\,w_2\,\delta_1;$$

and, dividing by $F\,\delta_1$,

$$m\,w_1 = w + n\,w_2\,w_1 = \frac{w + n\,w_2}{m}$$

By substitution for w_1, in the equation (1), we find

$$V = p_0 - p_1 = 2\,\delta_1\left\{\frac{w^2}{2\,g\,.\,m} - \frac{w^2}{2\,g\,m^2} - \frac{n^2\,w_2^2}{2\,g\,m^2} - \right.$$
$$\left. - \frac{2\,n\,w\,w_2}{2\,g\,m^2}\right\} \text{ ....(2)}$$

The density of the mixture of steam and air being very near that of the atmosphere itself, δ_1 may be allowed $= 1$.

$\frac{w^2}{2\,g}$ has been shown to represent the difference of pressures in the tube and in the chamber C—viz.,

$$\frac{w^2}{2\,g} = p - p_1 = x.$$

In a similar manner, $\frac{w_1^2}{2\,g} = x - p = p_1$ represents the pressure of the atmosphere above that within the chamber, which is equal to the vacuum $=$ V.

The value of $\frac{2\,n\,w\,w_2}{2\,g\,m^2}$ is very small, and may be neglected. Therefore we find

$$V = \frac{2\,x}{m} - \frac{2\,x}{m^2} - \frac{2\,n^2\,x}{m^2} + \frac{2\,n^2\,p}{m^2} = x - p;$$

$$\text{from which } x = \frac{(m^2 + 2\,n^2)\,p}{m^2 + 2\,(m-1) + 2\,n^2}; \;\ldots (3.)$$

$$\text{and the vacuum } V = \frac{2\,(m-1)\,p}{m^2 - 2\,(m-1) + 2\,n^2} \;\ldots (4.)$$

The formula (3) gives the vacuum as a function of the steam pressure p, and of the proportionate sizes of the air passage and chimney to the blast orifice, n and m. It shows that *the vacuum is in direct proportion to the blast pressure; that it decreases, although not in direct proportion, with an increase of the area of chimney for a given blast orifice; and decreases in a ratio corresponding to the square of the increase of air passage, although not in direct proportion with that square.*

The vacuum being known, it is very easy to establish the formula for the quantity of air, M_1, drawn into the chamber by the blast, and also the quantity of steam, M, required for creating that draft.

$$M_1 \quad F_2\,w_2 = F_2\,\sqrt{2\,g\,(x-p)};$$

and since $F_2 = n\,F$, it follows—

$$M_1 = n\,F\,\sqrt{2\,g}\,V.$$

By substitution for V, from formula (3), taking $g = 32$ feet per second, we have the value for M_1 in cubic feet—

$$M_1 = 8\,F\,n\,\sqrt{\frac{2\,(m-1)\,p}{m^2 - 2\,(m-1) + 2\,n^2}} \quad (5.)$$

The quantity of steam is in cubic feet per second,

$$M = F\,\sqrt{2\,g\,x} = 8\,F\,\sqrt{\frac{(m^2 + 2\,n^2)\,p}{m^2 - 2\,(m-1) + 2\,n^2}}. \;(6)$$

Therefore the proportion between air and steam,

$$\frac{M_1}{M} = n\,\sqrt{\frac{2\,(m+1)}{m + 2\,n^2}} = \sqrt{\frac{2\,n^2\,(m-1)}{m^2 + 2\,n^2}} \;\ldots (7.)$$

The equation (5) shows the manner in which the quantity of air drawn in by the blast depends upon the steam pressure and proportions of openings. With a given area of blast orifice, the letter m represents the sectional area of chimney, and n the flue-way or tube area. *The quantity of air is proportionate, therefore, to the square root of the steam pressure. It increases in a direct ratio with the area of flue-way, although it is not proportionate to the same, the relation being of a more complex character. The quantity of air decreases with an increase of chimney area.*

From (6) we find the relative quantities of air which a given weight of steam is capable of drawing through the chimney with given proportions of blast orifice and flue-way. The formula shows this quantity to be *independent of the steam pressure and of the absolute size of the blast orifice, and only affected by a variation in the proportions of the openings for steam and air.* If the flue-way, n, is much larger than the chimney, m, then the quantity of air set in motion by a given weight of steam will be increasing in a direct ratio almost with the flue-way. If, on the other hand, the chimney is wide, and the flue-way restricted, then the quantity of air will decrease nearly in proportion to the increase in chimney section.

Hitherto the effects of friction, and other drawbacks from contraction of the liquid vein, and from similar causes, have been neglected. This is admissible so long as the formulas are to be used only to show the proportionate quantities of vacuum, air, and steam under varied conditions. If, however, the formulas are to be used for calculating actual quantities, these resistances must be taken into consideration.

Call K the co-efficient of resistance offered to the steam in passing through the pipe A, so that its speed, w, be in consequence of that resistance changed into K w. The resistance offered by the pipe D to the passage of the air will not be materially different, circumstances being much the same in both cases. The speed w_2 will therefore be changed into K w_2. By making these substitutions in the equation (2), the expressions for V and M, from (4) and (5), will be transformed into the following:—

$$\text{Vacuum, } V = \frac{2\,(m-1)\,p}{k\,(m^2 + 2\,n^2) - 2\,(m-1)} \;\ldots (8.)$$

$$\left.\begin{array}{l}\text{Quantity of air, } M_1 = 8\,F\,n \\ \sqrt{\dfrac{2\,(m-1)\,p}{K\,(m^2 + 2\,n^2) - 2\,(m-1)}}\end{array}\right\} \ldots\ldots\ldots (9.)$$

The exact value of K has been ascertained in a series of experiments made in Zurich by Professor Zeuner.* The following table shows the value of K for different blast pressures:—

Blast pressure on the square inch in pounds, above the atmosphere $= p$,	7·35℔.	14·7℔.	22·05℔.
Value of K,	1·463	1·588	1·671

To put the above formula (8) to a test, an example taken from practice may be calculated.

An engine described by Mr. D. K. Clark, and experimented upon by that authority with the vacuum gauge, had the following dimensions:—

No. 124.—Area of ferules, 2·3 square feet.
Area of chimney, 1·67 square feet.
Area of blast orifice = 10 square inches.

With a blast pressure of 9 inches of mercury, this engine gave a vacuum of 4 inches of water.†

In this instance the ratio of chimney to blast orifice is $m = 24$.

The ratio of ferule area to the blast orifice is equal to

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* *Das Locomotiven Blasrohr*, Zurich, 1863.

† D. K. Clark, *Railway Machinery*.

32; but the ferule area does not represent the opening for admission of air in the elementary apparatus, Fig. 200, since it is not in direct communication with the atmosphere. It has been ascertained by experimental researches that the vacuum in the fire box is about 50 per cent. of that in the smoke box.[*] The action of the ferule area must therefore be reduced in proportion, and it may be considered equal to an opening 50 per cent. less in size, leading direct into the atmosphere. We have therefore $n = 16$.

The blast pressure, $p, = 9$ inches of mercury, is about 5 pounds to the square inch; therefore $k = 1{\cdot}4$, $p = 5$.

We find the vacuum—

$$V = \frac{2 \times 23 \times 5}{1{\cdot}4\,(16^2 \times 2 + 24^2) - 2 \times 23} = \frac{230}{1{\cdot}4 \times 1088 = 46}$$

$$V = 230 : 1477 = 0{\cdot}16 \text{ pounds per square inch.}$$

To find the vacuum in inches of water, we have the pressure per square inch equal to one inch of water $= 0{\cdot}036$ pounds; therefore the vacuum

$$V = \frac{0{\cdot}16}{0{\cdot}036} = 4{\cdot}4 \text{ inches of water.}$$

This corresponds very closely with the vacuum found by experiment.

We may now try to calculate the quantity of air drawn through the fire box of the same engine, with the given blast pressure of 5 pounds per square inch, or area of blast orifice of 10 square inches, and the corresponding vacuum of 0·16 pounds per square inch.

We have from formula M—

$$M = 8 \cdot 16.\ \frac{10}{144} \sqrt{0{\cdot}16} \text{ in pounds weight per second,}$$

$$= \frac{512}{144} \text{ pounds;}$$

and the weight of air passing the grate per hour

$$= \frac{512}{144} \times 3600$$

$$M = 12800 \text{ pounds.}$$

Allowing 15 pounds of air for each pound of coke burnt on the grate,[*] we have

12800 : 15 = 853 pounds of coke

as the corresponding hourly consumption of coke on the grate, which, with an evaporation of 9 pounds of water per pound of coke, gives

853 = 9 × 7677 pounds of water;

or, in cubic feet, 7677 : 62·5 = 122 cubic feet of water. This also very well corresponds with the figures found by actual experiment with engines of similar sizes and proportions, as stated by Mr. Clark.

The considerations and deductions given above are not intended for direct practical use, since calculations of inches of vacuum are not made nor required in designing or constructing locomotive engines. Their purpose is to give a clear insight into the relative functions of all the elements which constitute the blast apparatus of a locomotive engine, and particularly to show how the resistance of the blast pipe, or the labour required for creating the blast, may be ascertained by calculation, and may be taken into consideration in proportioning the cylinders in which the effect of this labour appears in the form of back pressure. The action of steam in the cylinder is illustrated by indicator-diagrams in the chapters immediately following, in which the resistance of the blast-pipe is investigated in particular examples.

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[*] *Guide du Mecanicien*, Paris, 1851.

[*] See Chapter XIII., "On Combustion."

CHAPTER XVI.*

THE BEHAVIOUR OF STEAM IN THE CYLINDERS OF THE LOCOMOTIVE.

It has been seen that, conformably to the periodical and contemporaneous operations of the piston, the valve, and the steam, in the cylinder of the locomotive engine, four distinct events take place in consecutive order with respect to each end of the cylinder: first, the admission of the steam at or just before the beginning of the stroke; second, the cutting off or suppression of the steam; third, the release or exhaust of the steam; and finally, the lock-up or compression of the exhaust steam, prior to the opening of the port for admission. These four events together constitute the "distribution" for the cylinder; and their durations, measured in parts of the stroke, are the "periods of the distribution." By the aid of the indicator, which, as its name implies, is a sort of stethoscope for the observation of what transpires within the cylinder—a simple instrument for receiving and registering the tension of the steam—a minute and accurate picture of the operations within is transferred by pencil to paper, affording valuable, and indeed indispensable, data for the measurement of the power and efficiency of the steam in the cylinder. The following is an abstract of the writer's experimental analysis of the behaviour of steam in locomotive engines, based chiefly on observations made by him in the year 1850.†

The Indicator-Diagram—The Behaviour of Steam in the Cylinder.—In illustration of the function and utility of the indicator, by means of which most of the writer's observations were conducted, examples of indicator-diagrams, obtained by him from one of the cylinders of a locomotive, are illustrated in Fig. 203. The base line, A B, is the line of atmospheric pressure, and represents the stroke of the piston; and the rectangular space above it may be supposed to be the interior of the cylinder. The heavily-lined figure is a diagram of the indicated action of the steam, when the piston moved in the cylinder at an average slow speed of 40 feet per minute, and shows by its angularity how the steam is controlled by the valve, and the precise points of the stroke at which the changes of the distribution take place. The piston is represented as having started from the right-hand end of the cylinder under a uniform pressure of 61 lbs. steam above the atmosphere, traced from the upper right-hand corner, till it reaches the point of suppression. The admission being terminated, the period of expansion is commenced, the pressure declines as the piston advances before the expanding steam, and continues to do so till the piston reaches the point of release. At this point the piston enters on its third and last stage of progress toward the end of the steam-stroke. The steam, primarily admitted at 61 lbs. above the atmosphere, and attenuated to 23 lbs. pressure previously to being released, quickly discharges itself into the atmosphere, in virtue of its remaining elasticity, and is entirely evacuated before the end of the stroke, as indicated by the quick and total decline of the steam line, during the period of exhaust, towards the point B. The evacuation is, however, only relative, not absolute, as steam of atmospheric pressure remains in the cylinder, though not obviously sensible in the indicator-diagram. During the return-stroke, therefore, the valve ought to maintain the exhausted end of the cylinder continuously open, to allow the steam of one atmosphere to escape before the returning piston; and the benefit of this provision is proved by the diagram, in which it appears that, during the continuation of the exhaust, the steam of latent pressure remains at the zero point of the scale. At the instant of closing or compression, however, when there is no longer an exit for the latent steam before the piston, the diagram-line slopes upwards towards the right-hand side, and the steam is compressed against the end of the cylinder. While the volume of the compressed steam is being thus forcibly reduced, the density is increased, the pressure is raised, until the accumulation of back pressure so induced is merged into the superior pressure of the steam admitted by anticipation for the business of the next steam-stroke. The point at which this irruption of pressure occurs is indicated by the small compartment A in the figure, representing the "period of pre-admission;" instantly the pressure mounts to the maximum initial pressure of 61 lbs., in readiness for the succeeding steam-stroke.

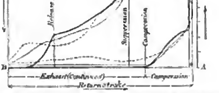

Fig. 203.—Indicator-Diagrams from a Locomotive Engine, to show the action of steam in the cylinders.

The behaviour of the steam in the cylinder may thus, with the aid of the indicator-diagram, be clearly traced through the cycle of changes. The period of admission, in the example just described, is, it appears, about one-third of the whole stroke; that of expansion is something more;

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* This and the subsequent portions of the Work have been contributed by Mr. D. K. Clark.

† See *Proceedings of the Institution of Mechanical Engineers*, 1852, for an account of the writer's observations on the "Expansive Working of Steam in Locomotives."

and a simple inspection of the diagram shows that, in this instance, one-half of the work of the steam is performed by simple expansion while shut up in the cylinder. Even the period of exhaust supplies its quota of effect, inasmuch as the evacuation is a work of time, and the extra positive pressure so yielded is represented by a small triangular space between the point of release and the end of the stroke. The force developed by compression is properly designated resistance, as it is opposed to the motion of the piston, and must be classed with the slight opposition also made by the entering steam during its pre-admission for the steam-stroke.

But the important inquiry remains to be urged,—How is the behaviour of the steam affected by the speed of the piston? If the piston move slowly, there is plenty of time for the steam to go through its mechanical duties. While the steam is admitted, it follows up the piston at full pressure; while the exhaust is open, it thoroughly evacuates itself. But at higher speeds, the time for each evolution is proportionally shortened; and it remains to be considered in what way this accelerated work is discharged. The dotted-line diagram (Fig. 203) illustrates the behaviour of the steam in the same cylinder, under the altered circumstance of a higher speed of piston, averaging 310 feet per minute, other circumstances being the same. The steam enters at an initial pressure of 62 lbs. per square inch, but suffers a slight reduction of pressure as the piston recedes before it,—a circumstance which may at once be attributed to the accelerating speed of the piston in the cylinder, specifically due to the nature of the crank-motion, and the consequently greater difficulty of following it. The difficulty, however, is not considerable; and it is only when the piston nears the point of suppression, and the port is nearly closed by the valve, that the pressure rapidly falls in the diagram towards the suppression-line. This is a case of simple wire-drawing, as the opening of the port, previously wide enough to admit all the steam that could find its way into the cylinder against the frictional resistance and bends of the passage, is now reduced to the minimum width consistent with this condition, and a further contraction and final closing necessarily occasions an accelerated fall of the pressure. The pressure at the instant of suppression, or cut-off, under these circumstances, is 54 lbs. above the atmosphere. The curve descends during the period of expansion, and cuts the line of release at a pressure of 19 lbs.; and on reaching the end of the stroke, it attains a minimum of 2 lbs. of pressure. The curve of expansion, it appears, runs into the curves of the admission and the exhaust without any of the abruptness which distinguishes the slow diagram; the fact being that, before the steam was nominally cut off, expansion, technically so called, had begun,—a result implied in wire-drawing; and there was, therefore, not the same liability to sudden change of pressure on entering the period of simple expansion. At the termination of the period of expansion, the curve crosses the exhaust-line nearly at right angles, and barely reaches the minimum pressure at the termination of the stroke. The comparative delay so evinced in the accomplishment of the exhaust, is plainly a consequence of the shorter time allowed for this purpose by the greater speed of the piston; and, accordingly, one perceives a material accession to the area of the diagram, or the useful effect of the steam, in the very circumstance that it exhausts less freely. On the other hand, a drawback on this additional effect exists in the sustained back pressure of 2 lbs. per square inch, as indicated during the return-stroke, referable to the same cause—imperfect exhaustion. The exhaust-line runs into the compression-line with a slight bend; and it is observable, that before the pre-admission takes place, the compressed steam attains to a higher pressure than that found by the slow diagram,—a circumstance to be explained by the condensation of the steam in the cylinder at the slow speed. But, though the curve of high speed is in advance of that of the slow speed at the instant of admission of fresh steam, it falls behind at the commencement of the stroke. At this point the pressure does not get beyond 51 lbs. above the atmosphere, and only attains the maximum, 62 lbs., when the piston has described half an inch of the steam-stroke. This deficiency is attributable chiefly to the shortness of time allowed for the re-establishment of the working pressure, and indicates the need for more lead of the valve.

In the flow of steam from the boiler to the cylinder, it meets with hindrances to its passage, which usually operate to cause a considerable reduction of pressure when it reaches the cylinder, even if all the passages be thrown wide open. The actual charge of steam transmitted through an irregular passage of considerable length, and of a given sectional area, is, in all cases, less than what can be passed through an aperture of a very short length, as in a plate of the same sectional area, owing to the bends and lateral friction of the long passage. It therefore frequently happens that the opening of the port allowed by the valve, though it may be much less than the total area of the port, is sufficiently large to pass all the steam that can force its way along the passage. This fact is constantly exemplified in practice. It is known that opening the port beyond a certain amount—in all cases less than the area of the port itself—ceases to be advantageous in facilitating the passage of the steam into the cylinder. Similarly, the opening of the regulator, or "throttle valve," beyond a small fraction of the sectional area of the steam pipe, does not add to the available pressure at the valve chest. When the steam is not dry—containing water in suspension—the labour of moving in passages is greatly increased, owing to the quantity of dead, inelastic weight to be dragged along; and the reduction of pressure is consequently much more than with dry steam.

Notwithstanding the objection that has been urged against the ordinary slide-valve, worked by an eccentric motion—insufficient celerity of motion—there is no material wire-drawing of steam by the closing valve when the period of admission exceeds two-thirds of the stroke, unless at very high speeds of piston, exceeding 500 to 600 feet per minute. When the steam is cut off at shorter periods, however, the travel of the valve being less, and therefore, also, its velocity of motion, the wire-drawing increases at high speeds.

CHAPTER XVII.

THE ACTION AND CAPABILITIES OF THE LINK-MOTION IN THE DISTRIBUTION OF STEAM.

THE action of the valve, in the "distribution" of the steam, is regulated by three elements—the lap, the lead, and the travel. When these are given, the point of the stroke of the piston at which the steam is admitted to the cylinder, cut off, exhausted, and compressed or shut up, are all deducible by means of models, by diagrams, or by calculation. This can be done whether the valve derives its motion from a single eccentric or from a link-motion, as the movement of the valve is virtually the same in both cases. The way in which the valve is caused to cut off or suppress the steam earlier by the link-motion, is by shortening the travel of the valve. This is accomplished by means of the reversing gear, in such a manner that, whatever be the reduction of travel communicated to the valve, the lead is always at least the same as in full gear, and, with the shifting link, it is rather increased. In shortening the travel, not only is the steam cut off at an earlier point of the stroke, it is also exhausted earlier, and admitted earlier, and the exhaust-port is closed earlier, during the return-stroke, upon the exhaust steam. Thus, by shortening the travel, everything affecting the distribution is done earlier in the course of the steam and the return-strokes.

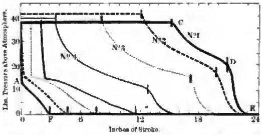


Fig. 204.—Indicator-Diagrams to illustrate the Distribution of Steam in the Cylinder by means of the Link-motion.

In his experiments on the action of steam, the writer employed M'Naught's indicator, which he applied to the front end of one of the cylinders of the engines; it received the alternate motion for the paper-cylinder from the end of the piston-rod, through an intermediate lever suspended from the engine-frame. To test the actual state of the valve-gear of each engine at the time of the experiment, so far as it affected the distribution, indicator-diagrams were obtained from the cylinder at very low speeds, under each notch of the sector of the reversing gear. The diagrams so obtained, of which some examples are given, are angular and sharply defined, and they show with precision at what points of the stroke the changes of the distribution take place. For instance, in the diagram, Fig. 204 (from a locomotive engine, fitted with shifting links, taken under full gear in the first notch of the sector, and marked No. 1), the opening of the port for the admission of steam commences at the point A, about $\frac{3}{16}$-inch before the beginning of the steam-stroke, when the line starts upward to the regular steam level at B, in time to commence the steam-stroke at full pressure. From B to C the steam is shown to be admitted to the cylinder at a uniform pressure of 38 lbs. At C, it is cut off, or suppressed; and while the piston moves from C to D, the inclosed steam expands behind it at a regularly decreasing pressure, shown by the curve C D. At D, the steam is exhausted, and the pressure quickly declines till the end of the steam-stroke, E. During the return, or exhaust-stroke, the steam continues to exhaust into the atmosphere, and the atmospheric line, E F, is described. When the piston arrives at F, the exhaust-steam is denied any further egress; and the piston continuing in motion, it compresses the steam against the end of the cylinder, and raises its pressure, as indicated by the line F A, until, at A, steam is admitted from the valve-chest for the next steam-stroke. The portions of the double stroke described by the piston during this succession of changes or events, traced for one face of the piston, are distinguished by the writer as the periods of the distribution; and the points of the stroke at which the changes occur, as the points of the distribution. These are further distinguished as follows:—

A is the point of admission.

C is the point of cutting off, or suppression.

D is the point of exhaust, or release.

F is the point of compression; also, the portion of the stroke described while the line A B is traced, is the period of pre-admission.

The portion of the stroke, B C, is the period of admission.

The portion between C and D is the period of expansion.

The portion between D and E is the period of exhaust during the steam-stroke.

The portion E F is the period of exhaust during the return-stroke.

The portion between F and A is strictly the period of compression; but the period of pre-admission is generally added to it, and thus the compression usually signifies the whole distance of the point, F, from the end of the return-stroke.

These definitions apply to diagrams taken from every notch of the sector, as it will be seen from the diagrams, Nos. 2, 3, 4, Fig. 204, from the same engine, that, however varied in form, they have all the parts of the diagram, No. 1, for full gear.

It is obvious from the diagrams, that the sooner the steam is cut off, the sooner it is exhausted, the sooner the port is closed for exhaustion, and the sooner the port is opened for the admission of steam. 2nd. That though every change is made earlier—as measured in parts of the stroke—there is less difference in the position of the points of exhaust, compression, and admission, than in that of the cutting off. Consequently, the shorter the admission the longer is the expansion, as the exhaust-point does not recede so much as the point of cutting off. 3rd. That by the shifting link-motion, the steam may be cut off at from $\frac{1}{8}$th to $\frac{3}{4}$th of the stroke. 4th. That though the exhaust takes place earlier for every increase of expansion, it does not in any case take place within the first half of the stroke. For mid-gear it occurs, in fact, at 54 per cent. of the stroke; and the steam is expanded into 3¼ times the length of stroke at which it is cut off. 5th. That the period of compression, increasing as the admission is reduced, amounts to about one-half stroke in mid-gear. 6th. That the pre-admission of the steam, which is not above 1 per cent. of the stroke in full gear, increases to about 10 per cent. in mid-gear.

These results are for an ordinary shifting link-motion, in every modification of which the lead increases with the degree of expansion, and in which the lead in this case rises from $\frac{4}{16}$ths to $\frac{9}{16}$ths inch in mid-gear. Whereas, in stationary link-motions, having the links suspended directly from a fixed point, the lead is constant for all degrees of expansion; and if in these the lead be set at ¼th to $\frac{5}{16}$th inch constant, we should be able to cut off at even 10 to 12 per cent. of the stroke, or at from $\frac{1}{10}$th to $\frac{1}{8}$th of the stroke. For example, in the diagrams from No. 125, Caledonian Railway Engine, in Fig. 205 (lap 1¼ inch, lead ¼ inch), the steam is shown, in No. 5 diagram, to be cut off at 3¼ out of 24 inches stroke, or at $\frac{1}{7}$th from the front end of the cylinder. Now, in this engine, as the valve-gearing was slightly out of balance, the steam was cut off 1 inch earlier for the back-stroke in mid-gear—that is, at 2¾ inches—and the mean of the two, or 3 inches, is the mean admission in mid-gear, or $\frac{1}{8}$th of the stroke.

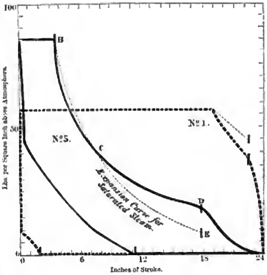


Fig. 205.—Indicator-Diagrams to illustrate the Condensation of Steam in exposed Cylinders.

It has been seen that the earlier the steam is cut off, the earlier also is it exhausted, until, in mid-gear, it may be released at half-stroke. This has been deemed a serious objection to the use of link-motions for high expansion, as it is supposed to lead to a serious loss of expansive action, by exhausting prematurely. This loss is, however, a mere trifle in practice. The escape of the steam is by no means instantaneous, as the slow diagrams in Figs. 203, 204, and 205, very clearly prove. Thus, in the diagram No. 1, Fig. 204, the exhaust-line, DE, shows that nearly all the period of

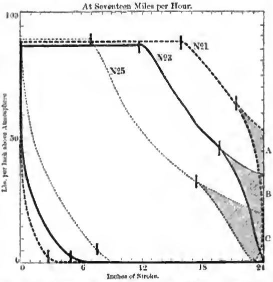


Fig. 206.—Indicator-Diagrams from the "Great Britain" Locomotive.

No. 1.	Mean Pressure,	Steam,	81·0lbs.	Exhaust,	2·5lbs.	Effective, 78·5lbs.
No. 3.	,,	,,	69·9 ,,	,,	3·8 ,,	,, 66·1 ,,
No. 5.	,,	,,	53·2 ,,	,,	9·7 ,,	,, 43·5 ,,

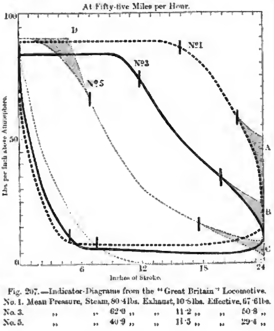


Fig. 207.—Indicator-Diagrams from the "Great Britain" Locomotive.

No. 1.	Mean Pressure,	Steam,	80·4lbs.	Exhaust,	10·8lbs.	Effective, 67·6lbs.
No. 3.	,,	,,	62·0 ,,	,,	11·2 ,,	,, 50·8 ,,
No. 5.	,,	,,	40·9 ,,	,,	11·5 ,,	,, 29·4 ,,

exhaust for the steam-stroke is employed for the complete evacuation of the steam. And, if this be the case for speeds of one and two miles an hour, it is much more so for the regular working speeds of trains. To select from a very admirable series of indicator-diagrams, with copies of which the writer has been favoured by Mr., now Sir Daniel Gooch, by whom they were taken from the cylinder of the "Great Britain" locomotive, on the Bristol and Exeter Railway, the Figs. 206 and 207 contain diagrams taken at seventeen and fifty-five miles per hour, respectively, under the 1st, 3rd, and 5th notches of the sector. The following are the conditions of the valve-motion of this engine when the diagrams were taken:—

STATE OF THE VALVES OF THE "GREAT BRITAIN," GREAT WESTERN RAILWAY.

Cylinder, 18 × 24 inches. Wheel, 8 feet.

Lap, ...	1¼ inch.	
Constant Lead, ...	⅜ ,,	
Travel in Full Gear, ...	4¾ ,,	
Blast orifice, ...	5⅛ ,,	diameter.

No. of Notch.	Position of Points of Distribution.			Period of Exhaust during the Steam-stroke.
	Cutting off.	Exhaust.	Compression.	
	Inches of Stroke.	Inches of Stroke.	Inches of Stroke.	Inches of Stroke.
1	16	21⅜	3	2⅝
3	11¼	19⅞	5	4⅛
5	7	17⅜	7½	6⅝

On the diagrams, the points of cutting off, and exhaust, are marked; and the steam-line falls only very gradually during the period of exhaust, especially at the high speeds. The expansion-curves are shown by dotted lines, A, B, C, Figs. 206, 207, continued to the end of the stroke. These are easily calculated in terms of the relative volumes of steam, from the pressures indicated at the points of exhaust, and are such as would have been described had the exhaust been delayed till the end of the stroke. The shaded areas, A, B, C, inclosed between these dotted curves and the curves actually described, express the power lost by exhausting the steam before the stroke is completed. Averaging them for the whole stroke, they are as follows:—

LOW SPEEDS, . .	1st Notch,	⅞ lb.	per inch	loss.
	3rd ,,	2¼ ,,	,,	
	5th ,,	3½ ,,	,,	
HIGH SPEEDS, . .	1st ,,	1 ,,	,,	
	3rd ,,	1 ,,	,,	
	5th ,,	⅞ ,,	,,	

The losses at high speeds are very small,—merely nominal; and, curiously enough, the loss by the earlier exhaust of the 5th notch is actually less than that under the 1st notch. The losses are, of course, greater at the low speeds; but even then, in the 1st notch, which is the only notch employed at very low speeds, the loss does not amount to 1 lb. per inch. The 3rd and 5th notches are employed only at speeds much above seventeen miles per hour, and the loss by them is of no practical moment.

Upon the whole, it follows that the possible loss by the early exhaust yielded by the link-motion is of no importance. On the contrary, it can be proved to be beneficial, as an early exhaust is at high speeds essential to a perfect exhaust during the return-stroke. It plainly appears, therefore, that with the existing arrangements of locomotives, any attempts to eke out the power on the steam-line, by prolonging the expansion materially beyond what is accomplished by an ordinary valve and link-motion, are not only useless, but highly prejudicial.

An objection has been made to the link-motion, that the steam is injuriously wire-drawn by it when under great expansion. Hence the numerous attempts that have been made to supersede the link by the employment of special expansion-valves. The diagrams may be referred to as examples of wire-drawing by the link. They were taken nearly consecutively with one opening of the regulator; and it is clear that the steam attained fully as high a pressure in the cylinder under the 5th notch as under the 1st. The pressure falls considerably towards the point of cutting off, and from the form of the steam-line, it is plain that very little additional steam is admitted for an inch or two before the cutting off actually takes place. The greater part of the steam is admitted at the higher pressure; and, in fact, a partial expansion of the steam already admitted takes place for some distance before the expansion nominally begins. Thus, the wire-drawing is, to a great extent, equivalent to an earlier cutting off, and a greater degree of expansion. The whole possible loss by wire-drawing is comprised within the dotted line, D, added to the diagram, which is merely an extension of the expansion-curve, to meet the steam-line, drawn horizontally to represent a free admission up to an imaginary point, D, of cutting off, 5 inches from the beginning of the stroke. This shaded area, D, amounts exactly to a mean loss upon the whole stroke of 1 lb. per square inch, by wire-drawing, under high expansion. For the 1st and 3rd notches, the amount of loss by wire-drawing must obviously be still less; and, in short, the objection of wire-drawing by the link-motion, when of liberal proportions, is of no practical weight.

Another objection to the link-motion, and apparently the most formidable one, is the large fraction of power neutralized by the compression of the exhaust steam, and which increases with the degree of expansion. Compression, however, involves no loss of efficiency; for, as by compression a quantity of steam is incidentally reserved and raised to a higher pressure, it gives out the power so expended in compressing it, during the next steam-stroke, just as a compressed spring would do in the recoil. But, apart from this general argument, the actual efficiency of the steam in the cylinder, with and without compression, may be exactly estimated. The most direct method of doing so, is to find the quantities of water consumed, as steam, for one stroke, under the two conditions, and to compare them with the relative effective mean pressures. It will suffice to analyze, as an example, the high-speed diagram under the 5th notch, No. 5. The volume of steam admitted is measured by the product of the area of piston (254·47 inches), and the period of admission, plus the total clearance in the cylinder and steam passage; the clearance being measured, for simplicity, in inches of stroke, we have $7 + 1·8 = 8·8$ inches, for the total volume admitted. The pressure of the steam when cut off is 65 lbs, for which the relative volume of water is 359.

Therefore the volume of water, as steam, or the water-equivalent of the steam admitted, is—

$$\frac{254{\cdot}47 \times 8{\cdot}8}{359} = 6{\cdot}24 \text{ cubic inches.}$$

From this is to be deducted the quantity of steam reserved by compression; the volume so reserved is measured by the period of compression, plus the clearance (7·5 + 1·8 = 9·3), and the pressure at the point of compression is 8 lbs., for which the relative volume is 1125. Then the water-equivalent of the reserved steam is—

$$\frac{254{\cdot}47 \times 9{\cdot}3}{1125} = 2{\cdot}10 \text{ cubic inches;}$$

subtracting, there remains 6·24 — 2·10 = 4·14 cubic inches of water as steam, actually expended for one stroke of the piston.

Were there to be no reservation of exhaust steam, by foreclosing the exhaust port, the whole area of resistance by compression would be removed, and there would be a reserve of steam of atmospheric pressure equal in volume to the clearance only. The relative volume of atmospheric steam is 1669, and the water-equivalent of the reserve would be—

$$\frac{254{\cdot}47 \times 1{\cdot}8}{1669} = 0{\cdot}27 \text{ cubic inches;}$$

the expenditure per stroke would be 6·24 — 0·27 = 5·97 inches of water. Now, the positive mean pressure during the steam-stroke, as indicated, is . 40·9 lbs. per inch.
And the mean resistance by compression is . . . . . . 11·5 lbs. „

Thus the effective mean pressure is . 29·4 lbs.

This effective mean pressure of 29·4 lbs. is maintained by a consumption of 4·14 inches of water per stroke; and it has just been found that, with the compression removed, the positive mean pressure of 40·9 lbs. per inch would be maintained by a consumption of 5·97 inches of water per stroke. The effective pressure created per cubic inch of water is therefore,

In actual practice, ... $\frac{29{\cdot}4}{4{\cdot}14} = 7{\cdot}1$ lbs.

And would be by removing compression, $\frac{40{\cdot}9}{5{\cdot}97} = 6{\cdot}9$ lbs.

These quantities are expressions of the relative efficiency of steam employed with and without compression: they are virtually identical, and show that the resistance by compression in the cylinder, due to the action of the link-motion, does not impair the efficiency of the steam.

The last objection to the use of the link requiring notice is, that, at high speed, considerable back exhaust pressure is created. The amount of this is very various; and it depends also on circumstances for which the link-motion is not responsible; such as a deficiency of inside lead (which is regulated by the lap), small ports, a small blast-orifice, and imperfect protection of the cylinder. It suffices, on the present occasion, to point to what can be done by superior arrangements, as exemplified in the diagrams from the "Great Britain." The cylinders of this engine are, in a manner, suspended in the smoke box, and thoroughly protected; the steam-ways are very large—13 × 2 inches—being in area about 1/13th of the cylinder; the exhaust passage is very direct, and the blast-orifice is 5¼ inches diameter, or about 1/11th of the area of cylinder. The diagrams prove that the per centages of back exhaust pressure, in terms of the positive mean pressure, at fifty-five miles per hour, are—

For the 1st notch, ... 8¼ per cent.
„ 3rd do. . . . . . . 5¾ „
„ 5th do. . . . . . Nothing.

Better results than these should not, in practice, be required; for when locomotives are adapted to their work, and running at high speeds, they ought not to require an admission of steam above half-stroke. However, the area of blast-orifice rules the back exhaust pressure; and when the cylinder is duly proportioned to the boiler, it is quite practicable, by a few modifications in detail, still further to increase the orifice sufficiently to banish all traces of back pressure of exhaust at all practicable speeds.

Relative Pressures of Steam in the Boiler, the Valve-chest, and the Cylinder.—As in the cylinder, so in the intermediate thoroughfares between that and the boiler, the movements of steam are affected by the conditions of dryness, as well as by the size and form of the passages. In well-protected cylinders, with dry steam, and ports 1/12th of the area of the piston, the fall of pressure, or wire-drawing, in the cylinder passages, is about 16 per cent. of the pressure in the valve-chest, at a speed of piston of 600 feet per minute; and with a port 1/10th the area of the piston, the fall of pressure does not exceed 10 per cent. at any speed. An example of the relative indicated pressures in the valve-chest and the cylinder of a locomotive is shown in Fig. 208, where the former is represented to have oscillated above 5 lbs. per square inch.

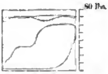


Fig. 208.—Relative Pressures in the Valve-chest and the Cylinder of a Locomotive Engine.

In imperfectly-protected cylinders, where the steam is not dry, the fall of pressure varies from 20 to 40 per cent. The fall of pressure in passing through pipes less than 1/15th of the piston in area, may be from one-third to one-fourth; in a pipe of 1/10th the piston, the fall is inconsiderable. If the steam be highly dried, a pipe of ordinary length, 1/15th the area of piston, is sufficient to transmit the steam undiminished in pressure.

The greatest useful opening of the regulator does not exceed 1/25th the area of the piston. A greater opening is not found to add to the facility of transmission.

CHAPTER XVIII.

OF THE EFFICIENCY OF STEAM WORKED EXPANSIVELY IN THE LOCOMOTIVE BY THE LINK-MOTION.

It is customary to apply the law of expansion discovered by Boyle, and better known as Marriotte's law, to determine the work done by steam acting expansively. In the present case, this mode of inquiry would be of little service; for though steam in well-protected cylinders expands nearly according to Boyle's law, or such that the total volume by expansion varies inversely as the total pressure, yet the results are affected by other circumstances,—chiefly the amount of clearance, wire-drawing, and back pressure of exhaust and compression. It will be preferable to take aggregate results of all these influences, as embodied in the model diagrams from the "Great Britain." Twenty-six indicator-diagrams were obtained, at speeds varying from fifteen to fifty-six miles per hour, of which the samples, Figs. 206, 207, taken at the opposite extremes of speed, suffice to point out the general characteristics. The annexed table contains, in the first nine columns, an analysis of these diagrams, which requires no further explanation. The effective horse-powers, column 10, are estimated in terms of the diameter and stroke of the cylinder, the diameter of the wheel, and the effective mean

RESULTS FROM INDICATOR-DIAGRAMS TAKEN FROM THE "GREAT BRITAIN."

No. of Diagram.	Speed of Engine in Miles per Hour	Indicated Steam Pressures in Cylinders in Lbs. per Square Inch.							Effective Horse-power Indicated.	Water-Equivalents.						Coke consumed per Effective Horse-power per Hour, allowing 1 lb. for 8 lbs. Water.
		Maximum Pressure during Admission.	Positive Mean Pressure.	Back Pressures.				Effective Mean Pressure.		Total admitted for One Stroke measured from Diagram.	Reserved by Compression.	Actually expended during One Steam-stroke.	Actually expended per Hour.	Actually expended per Effective Horse-power per Hour.		
				Exhaust.	Compression.	Sum of Back Pressures.										
						Pressure.	Per Cent. of Positive Steam Pressure.									
	Miles.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Per Cent.	Lbs.	H.-P.	Cub. Ins.	Cub. Ins.	Cub. Ins.	Cub. Ft.	Cub. Ins.	Lbs.	Lbs.
1	15	70	63·8	1·6	2·4	4·0	6·1	59·8	190	13·32	1·00	12·53	89·83	817	29·5	3·69
2	17	88	80·0	0·6	1·9	2·5	3·0	77·5	284	15·89	0·82	15·07	124·53	758	27·4	3·42
3	21	95	86·2	1·2	3·0	4·2	4·7	82·0	372	16·71	1·09	15·62	159·45	741	26·8	3·35
4	24	85	76·7	0·9	1·6	2·5	3·1	74·2	384	15·30	0·82	14·48	168·95	760	27·5	3·41
5	27	80	70·6	1·5	2·2	3·7	5·3	66·9	389	13·89	0·91	12·98	170·37	757	27·4	3·42
6	31	90	79·6	1·7	3·7	5·4	6·7	74·2	497	15·62	1·13	14·49	218·35	759	27·4	3·42
7	31	80	73·2	2·9	2·2	5·1	6·9	68·1	456	14·76	1·00	13·76	207·35	786	28·4	3·55
8	49	60	51·4	3·6	4·4	8·0	15·5	43·4	459	10·73	1·34	9·39	223·60	842	30·4	3·80
9	54	89	80·4	6·8	6·0	12·8	15·8	67·6	763	15·62	1·56	14·06	369·07	836	30·2	3·77
1st Notch—Means 82 lbs.								68·2							28·3	3·54
10	17	88	69·9	0·0	3·8	3·8	5·4	66·1	242	12·19	1·10	11·09	91·65	654	23·7	2·96
11	18	70	55·3	0·8	4·5	5·3	9·4	50·0	194	10·33	1·35	8·98	78·57	700	25·3	3·16
12	21	92	72·3	0·0	4·2	4·2	5·7	68·1	309	12·50	1·16	11·34	115·78	647	23·4	2·92
13	26	72	57·1	0·0	4·9	4·9	8·5	52·2	293	10·55	1·41	9·14	115·52	681	24·6	3·07
14	31	79	60·3	1·2	6·0	7·2	11·9	53·1	356	10·87	1·48	9·39	141·50	687	24·8	3·10
15	32	86	64·4	0·8	4·9	5·7	8·4	58·7	407	11·33	1·29	10·04	156·20	663	24·0	3·00
16	40	76	55·7	0·4	4·7	5·1	9·1	50·6	437	9·78	1·35	8·43	163·87	648	23·4	2·92
17	51	70	49·1	2·0	6·2	8·2	16·6	40·9	450	8·65	1·54	7·11	176·30	677	21·5	3·06
18	55	81	62·0	3·6	7·6	11·2	18·0	50·8	603	10·54	1·66	8·88	237·42	680	24·6	3·07
3rd Notch—Means 80 lbs.								54·5							24·3	3·03
19	17	89	53·2	0·0	9·6	9·6	18·0	43·5	159	7·90	1·85	6·05	50·00	543	19·6	2·45
20	18	70	42·1	0·5	6·6	7·1	16·7	35·0	136	6·59	1·76	4·83	42·26	537	19·4	2·42
21	21	93	56·5	0·0	6·3	6·3	11·1	50·2	228	8·20	1·42	6·78	69·21	525	18·9	2·36
22	28	74	41·8	0·4	6·2	6·6	15·7	35·2	213	6·87	1·68	5·19	70·64	537	20·7	2·59
23	31	88	46·5	0·0	7·4	7·4	15·5	39·1	262	7·29	1·59	5·70	85·89	566	20·5	2·56
24	36	80	39·0	0·0	8·5	8·5	21·1	30·5	237	6·08	1·85	4·23	74·02	540	19·5	2·44
25	50	77	34·7	0·5	8·0	8·5	24·4	26·2	263	5·52	1·76	3·76	91·39	600	21·7	2·71
26	56	90	40·9	0·0	11·5	11·5	28·1	29·4	353	6·24	2·10	4·14	113·20	554	20·1	2·51
5th Notch—Means 82 lbs.								36·1							20·1	2·51
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17

pressure in the ninth column. The water-equivalents, columns 11, 12, and 13, are estimated from the indicated pressure, and the period of the distribution for each notch, in the way already exemplified. The expenditure of steam per hour, column 14, is deduced from column 13, in terms of the speed, the cylinder, and the wheel; and dividing that by the effective horse-power, we have the contents of column 15, in inches, and of column 16, in lbs. Column 17 contains the coke, or coal, consumed per horse-power per hour, deduced for the several diagrams from the consumption of water, column 16, allowing 1 lb. of fuel to evaporate 8 lbs. of water.

Referring to the contents of the last two columns of this table, it is obvious that the consumption of water as steam, or of fuel, for a given amount of work done, becomes less the more expansively the steam is worked, and the means of the several quantities for the notches separately are as follows:—

CONSUMPTION PER HORSE-POWER PER HOUR.

For the	1st	notch,	28·3	lbs. water, or	3·54	lbs. coke.
„	3rd	„	24·3	„	3·03	„
„	5th	„	20·1	„	2·51	„

As the results under each notch vary very little, the means above stated may be adopted for all practical speeds without material error. To find from these means a formula which shall express the rate of economy by expansive working, it may be done graphically thus:—Draw a base line, A B, Fig. 209, to represent the stroke of the piston; set off on this base line the distances, A E, A F, A G, equal respectively to the periods of admissions under the 1st, 3rd, and 5th notches. From the points, E, F, G, draw the perpendiculars, equal respectively to the lbs. of water per horse-power per hour, consumed under the different notches, by any convenient scale, and terminate them by points as drawn; these points are found to range in a straight line, C D, which meet the vertical from A at a height of 14 lbs. by the scale, and the vertical from B at 36 lbs. The straightness of the line C D implies that the consumption decreases uniformly with the period of admission of the steam; the difference of height (36 — 14), or 22 lbs., is the whole decrease for the whole stroke. Consequently, if 22 be multiplied by the period of admission, and divided by the length of stroke, and 14 added to the quotient, the sum will express the consumption.

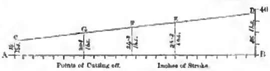


Fig. 209.—Diagram to show Consumption of Steam per Horse-power per Hour.

Let L = the length of stroke,

S = the period of admission of steam,

and W = the consumption of water in lbs. per horse-power;

then $W = 22 \frac{S}{L} + 14$, . . (1.)

or, at length:—

RULE I.—To find the consumption of water, as steam, per horse-power per hour, for a given period of admission. Multiply the period of admission in inches by 22, and divide by the length of stroke in inches, and add 14 to the quotient. The sum is the required consumption in lbs.

For the consumption of fuel, allowing 1 lb. for the evaporation of 8 lbs. of water, divide the water as above found by 8; and making C the consumption of fuel, we have—

$$C = 2{\cdot}75 \frac{S}{L} + 1{\cdot}75, \quad . \quad . \quad . \quad . \quad . \quad (2)$$

or, at length:—

RULE II.—To find the consumption of fuel, per horse-power per hour, for a given period of admission. Multiply the period of admission in inches by 2·75, and divide by the length of stroke in inches, and add 1·75 to the quotient. The sum is the consumption in lbs., per horse-power per hour.

These rules may be employed with safety for all periods of admission between 10 and 75 per cent. of the stroke which are the utmost limits worth regarding in the locomotive engine. They are applicable, also, for maximum pressures during admission, ranging between 60 lbs. and 120 lbs., though based on results from steam of 80 lbs. to 84 lbs. maximum pressure. For extreme pressures, the results, by the rule, are slightly too small in the case of lower pressures, and rather greater for the higher,—these divergences being due to the constant deduction of 15 lbs. for atmospheric resistance from the total pressure. For pressures above 80 lbs. the rules are perfectly safe, as they err rather by excess, on the safe side. The relative advantage of higher pressures, in respect of the constant loss by the atmosphere, progresses but slowly for pressures above 80 lbs. At this pressure, or 95 lbs. total, the atmosphere deducts $\frac{1}{6\cdot33}$rd; and at 100 lbs., the loss is $\frac{1}{7\cdot66}$th. The difference of these fractions is $\frac{1}{36}$th, which is all the economy, on account of atmospheric resistance by the use of 100 lbs. steam, compared with the work done at 80 lbs. For 120 lbs., the economy is $\frac{1}{21}$, and for 150 lbs., it is $\frac{1}{15}$th, with respect to a pressure of 80 lbs. The chief advantage, therefore, of the highest pressure, is in the liberty of working more expansively, while developing power at a given rate.

The following table shows, in the second column, the consumption of steam worked expansively, per horse-power per hour, due, by Rule I., to the periods of admission named in the first column, and expressed in percentage of the stroke. The inverse ratios of those quantities of steam are entered in column 3, a consumption of 100 per cent. being expressed by 1. Thus the actual relative efficiency of steam is found for various admissions. The fourth column contains the theoretical maximum relative efficiency of steam, expanding to the end of the stroke, according to the law of Boyle, with a perfect vacuum behind the piston, and without clearance, back pressure or compression, extracted from the ordinary tables on the subject. In column 5 are given the relative amounts of work done by steam, under the admissions named in column 1, being directly as the effective mean pressures in the cylinder, which are found by a rule to be afterwards given.

EFFICIENCY OF STEAM BY EXPANSION IN THE CYLINDER OF THE LOCOMOTIVE IN ACTUAL PRACTICE.

FOR MAXIMUM PRESSURES DURING ADMISSION OF 60 LBS. TO 120 LBS.

Periods of Admission in Parts of Stroke.	Water as Steam consumed in Lbs. per H.-P. per Hour.	Relative Efficiency of Steam in Actual Practice.	Possible Maximum Efficiency.	Relative Work done by Steam of the same Maximum Pressure in the Cylinder.
Per Cent.	Lbs.			Lbs.
10	16·2	2·22	3·30	15
12·5	16·7	2·15	3·08	20
15	17·3	2·08	2·90	24
17·5	17·8	2·02	2·73	28
20	18·4	1·96	2·60	32
25	19·5	1·85	2·39	40
30	20·6	1·75	2·20	46
35	21·7	1·66	2·05	52
40	22·8	1·58	1·92	57
45	23·9	1·50	1·80	62
50	25·0	1·44	1·69	67
55	26·1	1·38	1·60	72
60	27·2	1·32	1·51	77
65	28·3	1·27	1·43	81
70	29·1	1·23	1·35	85
75	30·5	1·18	1·28	89
100	36·0	1·00	1·00	100

From this table it appears that, in actual practice, the relative efficiency of steam, when cut off at one-tenth of the stroke, is 2¼ times greater than when not cut off till the end of the stroke, but that theoretically the increase should be 3¼ times, instead of only 2¼; therefore the actual efficiency of steam increases with expansive working, at a much slower rate than would be possible if every drawback were extinguished. Atmospheric resistance, it is expected, may be removed; and a material advantage would result from a reduction of the clearance between the valve and the piston.

As 75 per cent. is the greatest degree of admission really required with the link-motion; the relative efficiency for that period of admission (1·18) being compared with the efficiency (2·22) for 10 per cent. of admission, they are as 1 to 1·9, or nearly 1 to 2; and it follows that, under the most favourable existing circumstances, the utmost possible efficiency of steam worked expansively in locomotive engines by the link-motion is about twice that of the steam when worked under full gear; that is, the same quantity of steam does twice the quantity of work.

The effective mean pressure is to some extent affected by the speed; but, to find the average rate of increase with the admission, take the means of the maximum and of the effective mean pressures in the table, page 241, columns 3 and 9, as follows:—

No. of Notch.	Average Maximum Pressure.	Average Effective Mean Pressure in Cylinder.	
		Pressure.	Percentage of Average Maximum Pressure.
	Lbs.	Lbs.	Per Cent.
1	82	68·2	83
3	80	54·5	68
5	82	36·1	44

The percentages in the last column may be arranged in a curve, Fig. 210, having the base line, A B, to represent the stroke, and the perpendiculars at E, F, G, equal to the respective percentages measured on a vertical scale. The curve must pass through the point A, since, when no steam is admitted, no work can be done; the other end, at D, must also terminate somewhere below 100 by the scale, as, even with an entire admission, something must come off for back pressure. From the curve, the following formula is derived for finding the percentage of effective mean pressure due to a given admission, in terms of the maximum pressure:—

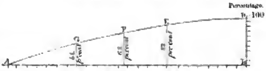


Fig. 210.—Diagram to show the Effective Mean Pressure due a given Admission.

Let A = the percentage of admission, and P = the percentage of effective mean pressure; then,

$$P = 13{\cdot}5\sqrt{A} - 28, \quad . \quad . \quad . \quad (3.)$$

RULE III.—To find the effective mean pressure in the cylinder, in terms of the maximum pressure, for a given percentage of admission. Multiply the square root of the percentage of admission by 13·5, and subtract 28 from the product; the remainder is the effective mean pressure in percentage of the maximum pressure of steam admitted.

EFFECTIVE MEAN PRESSURE IN THE CYLINDER, FOR VARIOUS ADMISSIONS.

Periods of Admission in Parts of the Stroke.	Effective Mean Pressure in Parts of Maximum Pressure.	Periods of Admission in Common Fractions of Stroke.	Effective Mean Pressure in Common Fractions of the Maximum Pressure.
Per Cent.	Per Cent.		
10	15	1-10th	• 1-7th full
12·5	20	...	...
15	24	1-8th	1-5th
17·5	28	...	...
20	32	1-6th	1-4th
25	40	1-5th	1-3rd
30	46	1-4th	1-2·5th
35	52	1-3rd	1-2nd
40	57	...	...
45	62	...	...
50	67	1-2nd	2-3rds
55	72	...	...
60	77	...	...
65	81	2-3rds	4-5ths
70	85	...	...
75	89	3-4ths	9-10ths
100	100	1	1

The results by this rule are rather too small for lower speeds, and rather too great for higher; but the deviations are of no practical moment. At forty miles per hour, or 560 feet of piston per minute for the "Great Britain," the result exactly coincides with practice; and this is an ordinary speed of piston in both goods and passenger engines, as, though the usual speed of the former engines on the rails is less than that of the latter, the wheel is smaller, and the stroke is commonly longer. The rule applies very well to admissions between 10 and 75 per cent., and to pressures (maximum) from 60 lbs. to 100 lbs., or even 150 lbs.

The preceding table of effective mean pressures is calculated by means of the foregoing rule, for admissions advancing by twentieths of the stroke, or intervals of 5 per cent.; and most of them may be accurately expressed in common fractions, as in the last column of the table.

In all well-protected cylinders, with blast-orifices not less than $\frac{1}{16}$th of the area of the cylinder, the foregoing rules and tables of data apply to the action of steam at speeds under thirty to forty miles an hour. For speeds amounting to fifty-five to sixty miles an hour, the loss by imperfect exhaust causes a large increase of consumption, per horse-power per hour, of from 33 to 12 per cent., according to the amount of admission. With steam-ports of about $\frac{1}{12}$th, and blast-orifices $\frac{1}{14}$th, of the cylinder, the rules likewise apply, at speeds under thirty to forty miles an hour. At the higher speeds, the useful power is considerably impaired by imperfect exhaust.

The proportions of the engines of the "Great Britain" may be applied to any other engine; and they may be repeated here as standard ratios for practice:—

Sectional Area of Cylinder,	. .	1
Do. Steam-port,	. .	1-10th.
Do. Blast-orifice,	. .	1-11th.

Lap of Valve, 1⅛ inch.
Travel, 4⅝ inches, in full gear.
Lead, ⅛ to ¼ inch.

There is, in general practice, a deficiency of accommodation for working steam so expansively as it could be worked with advantage, owing to the limited dimensions of the cylinders. The average period of admission of steam into the cylinders, even on light duty, is, in ordinary practice, as much as from 40 to 50 per cent. of the stroke, showing that the practice of expansive working in locomotives is open to improvement. There is no great difficulty in arranging, for general practice, to cut off at least as early as one-fifth of the stroke, which would economize 25 per cent. of the fuel consumed for the usual average period of admission of four-ninths of the stroke.

CHAPTER XIX.

OF THE CONDENSATION OF STEAM IN IMPERFECTLY-PROTECTED CYLINDERS, AND THE MEANS OF ITS PREVENTION.

The condensation of steam in the cylinder, by exposure, which takes place in certain arrangements of locomotives, is susceptible of proof in various ways: by the internal evidence of the indicator-diagram in respect of its general form, the form and course of the expansion-line, and the back pressure; also, by a comparison of the volume of sensible steam which is found to pass through the cylinder, with the volume of water found by measurement to be consumed from the tender and the boiler. The evidence of the expansion-line of the indicator-diagram will be first considered, both for well-protected and partially-protected cylinders.

Of the Evidence of the Expansion-line of the Indicator-diagram.—By Regnault's experiments, it is proved that the total heat of saturated steam increases slightly with the pressure, at such a rate that, for atmospheric steam, it is 1179° Fahr., and for 100 lbs. steam it is 1217°, or 38° more. This difference is of little importance, except as it shows that, when steam of a higher pressure is expanded, and falls to a lower pressure, it becomes slightly surcharged with heat as it expands,—assuming that it does not part with any of its heat, and that there is at least no necessary condensation of steam during expansion, and that, in fact, there cannot be any except what arises from the abstraction of the heat of the steam by work, or by external causes.

If water be present with the steam in the cylinder during expansion, as there commonly is, the heat of surcharge would convert a part of this water into steam. If 100 lbs. steam be expanded down to 20 lbs. or 30 lbs., the accession of steam in this way would be about $\frac{1}{40}$th of what is originally admitted, although the difference is so small as not to require further consideration in the present inquiry.

The slow diagrams (from No. 13 Caledonian Railway engine), Fig. 204, page 237, supply examples of the results of condensation, and its influence in modifying the expansion-line. If the steam which is cut off be permitted to expand in the cylinder, without any abstraction of its own heat, there can be no alteration of the whole quantity or mass of steam, whatever the change of volume may be; and the quantity of steam virtually saturated, indicated at the end of the expansion, or at any intermediate stage, should be found the same as at the commencement. If it be either greater or less, some change, by condensation or otherwise, must have taken place in the condition of the expanding steam. Referring to the diagrams from No. 13, if the whole clearance at the end of the cylinder, including the port, measured by 1½ inches of stroke, be added to the volumes of the steam at the beginning and end of the expansion, the sums so found will be measures of the total initial and final volumes of the steam expanded; and the ratios of these volumes for each notch are contained in the second column of the table annexed. The observed initial and final sensible pressures are added in columns 3 and 4, dividing, in each case, the total volume of the steam (equal to the product of the area of the piston and the length of the stroke representing the volume) by the relative volume due to the pressure; the quotient is the water-equivalent, or the volume of water at 60°, from which the steam is formed. The initial and final water-equivalents for each notch are entered in columns 5 and 6, and their differences in column 7, distinguished as positive (+), if in excess of the initial quantity, and as negative (—), if in deficiency; the eighth column contains the values of these differences as percentages of the initial equivalents. In column 9 are the pressures with which the expansion would have terminated had the initial quantity of steam in each case been preserved intact throughout the expansion. The pressures are found, in each case, by multiplying the relative volume of steam of the initial pressure by the ratios in column 2, which gives the relative volume of the final body of steam, and consequently its pressure. Column

OF THE EXPANSION AND WATER-EQUIVALENTS OF STEAM IN THE CYLINDER OF No. 13, C. R.

No. of Notch.	Ratio of Total Initial and Final Volumes.	Observed Pressures.		Water-Equivalents.				Final Pressure due to an equal Final Equivalent.	Difference of Final Pressures in Cols. 4 and 9.
		Initial	Final.	Initial.	Final.	Difference of Initial and Final.			
	Ratio	Lbs.	Lbs.	Cubic In.	Cubic In.	Cubic In.	Per Cent. of Initial.	Lbs.	Lbs.
1	1 to 1·34	38	22	4·73	4·56	—·17	— 3½	23½	+ 1½
2	1 to 1·58	41	19	3·99	4·01	+·02	+ ½	19	0
3	1 to 1·95	38	16	2·74	3·26	+·52	+ 19	10½	— 5½
4	1 to 2·66	39	13	1·58	2·29	+·71	+ 45	4	— 9
1	2	3	4	5	6	7	8	9	10

10 contains the difference of the final pressures so calculated, and those actually observed, column 11.

By tracing expansion-curves in dot-lines on the diagrams of Fig. 204, starting from the final pressures in column 9, and otherwise such as would have been described, with a constant quantity of saturated steam under expansion, the deviations of the actual curves from these, as standards, are easily shown, as in Fig. 211. For No. 1 diagram, the new dotted curve, C D, lies for its whole length above the actual curve, and terminates at 1½ lbs. more. For No. 2, the curves nearly coincide; for No. 3, the new curve proceeds for some distance above the actual, then crosses, and falls lower as it advances, until it ends at 5½ lbs. below the other; for No. 4, the new curve, A B, passes on as in No. 3, and ends at 9 lbs. below the actual. These deviations are all referable to one cause—the condensation of the steam.

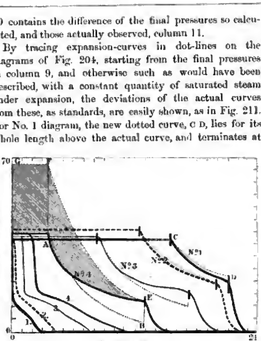


Fig. 211.—Indicator-Diagrams to illustrate the Condensation of Steam in the Cylinder.

In No. 1, the cylinder must have been at a lower temperature than the steam during the admission, and some condensation must have taken place; for no sooner is the steam cut off, than condensation is made visible, by the sinking of the expansion-curve below the standard, throughout the whole of its length. In No. 2, also, this takes place to a small extent for the first half of the curve, when the temperatures of the steam and the material of the cylinder become equal; after this, as the pressure continues to fall, and the temperature of the steam with it, the curve rises and meets the standard curve at the end, in virtue of a partial re-evaporation of the steam previously precipitated, caused by the cylinder itself, which, colder than the steam, and heated by it in the first stage of the expansion, is now relatively hotter, and partially restores the heat of which it had previously robbed the steam.

In Nos. 3 and 4, the process of successive condensation and re-evaporation is still more distinctly brought out. In these cases the greater portion of the heat engaged in the restoration of the steam during expansion must have been absorbed by the cylinder by condensation of the steam during admission. A reference to columns 7 and 8 shows the magnitude of this condensing agency; for, under the 3rd and 4th notches, the observed final equivalents are shown to exceed the initial by 19 and 45 per cent. of the latter respectively, which proves that, in the two cases, at least 19 and 45 per cent of the steam admitted must have been condensed during admission, as the additional steam can have been obtained from no other source. Although the actual expansion-curves, Nos. 3 and 4, indicate much higher mean pressures during expansion than the standard curves, and may, so far, be viewed as superior results, the favourable difference is only a partial amends for the much greater loss by initial condensation; and an expansion-curve may be constructed backwards, in terms of the indicated mass of steam at the end of the expansion, to show from what initial pressure this mass of steam could have expanded, had there been no condensation. Take No. 4, for example. The final pressure at E is 13 lbs., for which the relative volume is 939, and the ratio of the initial and final total volumes, or the degree of expansion, is 1 to 2·66; then 939 ÷ 2·66 = 353, which is the relative volume for 66½ lbs. steam at the point of suppression. Tracing the expansion-curve, E F, for this pressure, as in the drawing, for which any number of intermediate points may be found in the same way, and drawing a horizontal admission-line, F G, to the beginning of the stroke, the extra shaded area, so inclosed, is a representation of the real loss incurred by initial condensation of steam; and, without going into figures, it appears nearly as much again as the area, or power actually obtained.

The dotted curve, B C E, in Fig. 205, page 238, represents the proper expansion-line due to the initial pressure at B, in diagram No. 5; and, similarly, the dotted curve added in diagram No. 1, shows how the actual rate of expansion deviates from the proper rate.

The diagrams just discussed are, of course, extreme cases, which might occur in any cylinder, outside or inside; and they have been selected simply for purposes of illustration. They have served to show in what way the expansion-curves of indicator-diagrams may be turned to account in testing the condition of the steam. Our business is now to find to what extent, in the ordinary working of locomotives, the condition of the steam is affected by the circumstances of the cylinder.

It so happens, though not necessarily so, that inside cylinders are in general better protected than outside cylinders. The former are more completely within the smoke box and are more closely in contact with the smoke, and derive more benefit from its heat, than the latter; though, of course, there are many examples of inside cylinders being, for mechanical reasons, completely excluded from the smoke box, and having no other advantage over outsides than that they are less exposed to atmospheric draughts. The distinction of outside and inside must be understood to refer, not essentially to constructive arrangements, but to the incidental conditions of exposure or of protection.

The argument will be based on the results obtained for the well-protected cylinders of the "Great Britain," Great Western Railway, on the one hand, and the partially-protected cylinders of the Caledonian Railway passenger and goods engines, on the other, of which the passenger class, derived originally from the Crewe pattern, represent a widely-ramified species of outside-cylinder locomotive. Fig. 212 shows cross-sections of the cylinders and smoke boxes of the three classes of engines now referred to, in which it is apparent that the inside cylinders are the best protected.

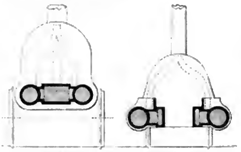

"Great Britain." Caledonian Railway.

Fig. 212.—Cross-Sections of the Cylinders and Smoke Boxes.

The first point is to show, by the expansion-line, that in well-protected cylinders the steam is not subject to condensation. Referring to the twenty-six indicator-diagrams from the "Great Britain," of which specimens were given in the last chapter, the following table contains, in column 1, the initial and final volumes of the expanded steam, clearance included, measured in inches of the stroke; in columns 3 and 4, the observed initial and final pressures; in columns 5 and 6, the initial and final water-equivalents of the expanded steam, deduced in terms of the capacity of the cylinder, and the relative volumes of steam due to the pressures; column 7 contains the difference of these equivalents. At the foot of the table are added the means for each notch.

It appears that for each notch the influence of speed on the relation of the initial and final water-equivalents of the steam expanded is nearly inappreciable. Dealing, therefore, with the means, it appears that the mean differences, column 7, constitute—

For the 1st notch, 3 per cent. of the initial equivalent.
" 3rd " 5½ " " "
" 5th " 2¼ " " "

These percentages are practically nothing; and the virtual constancy of the mass of expanding steam during expansion, thereby proved, shows that for the greatest observed degrees of expansion in the cylinder of the "Great Britain," no change in the condition of the steam is observable, and that there is, consequently, no condensation at all. Experiments made by the writer on some of the engines of the Edinburgh and Glasgow Railway, with inside cylinders, lead to the same conclusion.

OF THE EXPANSION AND WATER-EQUIVALENTS OF STEAM IN THE CYLINDER OF THE "GREAT BRITAIN."

Initial and Final Volumes by Expansion in Inches of the Stroke, Clearance Included.	No. of Diagram.	Observed Pressures during Expansion.		Equivalents of Water.		
		Initial.	Final.	Initial.	Final.	Difference.
		Lbs.	Lbs.	Cubic Inches.	Cubic Inches.	Cubic Inches.
1st Notch.	1	70	50	13·32	13·37	+ 0·05
	2	88	65	15·89	16·16	+ 0·27
Cutting off, 17·8 inches.	3	94	65	16·71	16·16	+ 0·55
	4	84	57	15·30	14·65	—0·65
Exhaust, 22·8 inches.	5	74	49	13·89	13·19	—0·70
	6	86	55	15·62	14·29	—1·33
Expansion, 1 to 1·3 inches.	7	80	56	14·76	14·47	—0·29
	8	52	34	10·73	10·32	—0·41
	9	86	60	15·62	15·23	—0·39
3rd Notch.	10	87	48	12·19	11·84	—0·35
	11	70	37	10·33	9·95	—0·38
Cutting off, 13·8 inches.	12	90	48	12·50	11·84	—0·66
	13	72	38	10·55	10·12	—0·43
Exhaust, 20·8 inches.	14	75	38	10·87	10·12	—0·75
	15	79	38	11·33	10·12	—1·21
Expansion, 1 to 1·5 inches.	16	65	33	9·78	9·24	—0·54
	17	55	28	8·65	8·35	—0·30
	18	72	37	10·54	9·95	—0·59
5th Notch.	19	89	33	7·90	8·30	+ 0·40
	20	70	24	6·59	6·90	+ 0·37
Cutting off, 8·8 inches.	21	93	33	8·20	8·35	+ 0·15
	22	74	21	6·87	6·42	—0·45
Exhaust, 18·8 inches.	23	80	22	7·29	6·58	—0·71
	24	63	17	6·08	5·76	—0·32
Expansion, 1 to 2·14 inches.	25	55	15	5·52	5·43	—0·09
	26	65	16	6·24	5·60	—0·64
Mean of 1st Notch, ...		79	54·60	14·65	14·21	—0·44
" 3rd "		74	38·40	10·75	10·17	—0·58
" 5th "		74	26·25	6·83	6·67	—0·16
1	2	3	4	5	6	7

Of the diagrams obtained from outside-cylinder engines of the Caledonian Railway, seventy-six were selected by the writer as average samples obtained by him during the regular work of the engines. These have been analyzed in the way adopted for those of the "Great Britain," and the mean results for each engine are tabulated below, ranging from 9 per cent. deficiency, to as much as 67 per cent. excess at the greatest expansion. Specimens of the diagrams from No. 42 passenger engine, and from No. 125 goods engine, are given in Fig. 213. These diagrams were taken by means of M'Naught's indicator; and the dotted lines show the actual curves, which are affected by the oscillation to which that indicator is subject at high velocities. The mean lines have been drawn on the diagrams on the principle which the writer has satisfied himself applies in the particular case of the indicator,—that action and reaction are equal, and that therefore the mean line, or radical form, ought to inclose the same collective area of diagram as the fluctuations in the lines actually described,—due partially to momentum, cutting off at one place as much as it incloses at another.

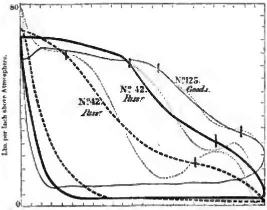


Fig. 213.—Indicator Diagrams at Quick Speed from Nos. 42 and 125, Caledonian Railway.

From this it appears that, for the greater ratios of expansion, the final equivalent of the steam is much above the initial, and the greater the ratio the greater is the percentage of this excess—amounting to 67 per cent., with an expansion of 3½ times. This relation is just what was found for the slow diagrams from No. 13; and there is no doubt that the excess of steam at the termination of the expansion is due to the same cause—namely, the condensation of steam in the cylinder during admission, and during the first part of the expansion, and the subsequent re-evaporation of a portion of the precipitated steam. During the experiments, there was at all times ocular demonstration of the existence of water in the cylinder, in the spray which escaped from it through the indicator, and which was given off more abundantly the more expansively the steam was worked.

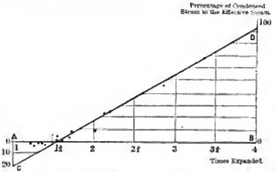


Fig. 214.—Diagram of the Condensation of Steam in Outside-Cylinder Engines, with the different Degrees of Expansion.

To find the general rate at which the percentage of condensation increases in these engines with the degree of expansion, the results obtained above may be referred, as ordinates, to a base line representing the ratios of expansion. Let A B, Fig. 214, be a base line divided to represent the total volumes by expansion, in terms of

OF THE EXPANSION AND WATER-EQUIVALENTS OF STEAM IN THE OUTSIDE-CYLINDER ENGINES OF THE CALEDONIAN RAILWAY; ABSTRACTED FROM THE RESULTS OF SEVENTY-SIX INDICATOR-DIAGRAMS, OBTAINED IN 1850.

No. of Engine.	Cylinder.		No. of Notch.	Initial and Final Volumes by Expansion, Clearance included, in Inches of Stroke.		Ratio of Initial and Final Volumes.	Observed Pressures during Expansion.		Equivalents of Water.			
	Diameter	Stroke		Initial.	Final.		Initial.	Final.	Initial.	Final.	Differences.	Ratio of Differences to Initial Equivalent.
No.	Inches.	Inches.	No.	Cub. Ins.	Cub. Ins.	Ratio.	Lbs.	Lbs.	Cub. Ins.	Cub. Ins.	Cub. Ins.	Per Cent.
13	15	20	1	14·02	18·82	1·35	52	34	5·93	5·92	— 0·01	— 0·2
„	...	...	2	11·27	17·82	1·60	45	24	4·26	4·51	+ 0·25	+ 1·2
33	15	20	4	13·33	18·75	1·40	45	22	5·02	4·57	— 0·45	— 9·0
„	...	...	5	12·00	18·30	1·52	56	27	5·27	4·99	— 0·28	— 5·3
„	...	...	7	8·00	16·10	2·01	46	16½	3·12	3·38	+ 0·26	+ 8·3
41	15	20	1	15·50	19·60	1·26	36	24	5·07	5·00	— 0·07	— 1·4
„	...	...	2	14·20	18·60	1·31	26	16	3·84	3·82	— 0·02	— 0·5
42	15	20	3	12·10	18·35	1·52	55	30	5·27	5·34	+ 0·07	+ 1·3
„	...	...	4	10·10	17·10	1·70	49	22	4·06	4·16	+ 0·10	+ 2·4
„	...	...	5	4·50	15·10	3·33	68	22	2·29	3·68	+ 1·39	+ 67·0
51	15	20	3	10·36	17·70	1·72	59	29	4·56	4·94	+ 0·38	+ 8·3
„	...	...	4	7·10	15·50	2·20	50	20	2·91	3·63	+ 0·72	+ 25·0
125	17	24	2	18·80	23·80	1·27	53	20	7·48	7·20	— 0·28	— 3·7
„	...	...	3	14·90	23·00	1·54	48	26½	7·57	7·98	+ 0·41	+ 5·4
„	...	...	4	10·20	22·10	2·16	37	13½	4·36	5·39	+ 1·03	+ 24·0
127	17	24	5	5·70	16·20	2·84	57	20	3·26	4·77	+ 1·51	+ 46·3
1	2	3	4	5	6	7	8	9	10	11	12	13

the initial volumes; and from B draw the vertical scale to measure the relative percentages of condensation. From A set off on the base line the ratios of expansion, and for each ratio set off perpendicular distances by the vertical scale, equal to the respective percentages of the differences of water-equivalents, column 13, and define their extremities by points, setting off minus percentages below the line, and plus percentages above. The mean line, C D, drawn through these points, is straight, and represents the mean rate at which the indicated condensation increases with the degree of expansion. It is found to meet the vertical from division 1, at 20 per cent. below; it crosses the base line at a volume of 1·53, and terminates at E, the point due to an expanded volume of 3·4, and to a percentage of 70, and would, if produced, meet the vertical from B at 92½ per cent. The straightness of the line implies that the indicated percentage of condensation increases uniformly with the relative volume by expansion. For an expansion of 1·53 times, the percentage of condensation, or indicated difference of equivalents, is nothing; and, generally, for expansions advancing by half-volumes, the percentages are as follows:—

Expanded Volumes, the Initial Volume being = 1.	Indicated Percentages of Condensation.
1·5	− 1¼
1·53	0
2	17½
2·5	36¼
3	55
3·5	73¾
4	92½

For every half-volume of expansion there is an increase of 18¾ per cent. of indicated condensation; and this becomes so serious, that, for an expansion of four times, if this were practicable with ordinary valves and link-motions, there would be 92½ per cent. of loss by condensation, or a loss of nearly one-half of the total quantity of steam admitted.

For ready reference, it is expedient to find the relative expansion and indicated condensation for different periods of admission, yielded by ordinary link-motions. The following table contains, in column 2, the total expanded volumes due by the nature of link-motion to the several periods of admission in column 1; and column 3 contains the relative indicated percentages of condensation due to these expansions, measured from the diagram.

Though the losses shown in the third column are great, the real losses must be still greater; because the restoration of condensed steam, by which the losses have been measured, cannot be complete. The indications, indeed, fail to show any loss at all at 50 per cent., for the re-evaporation balances the condensation during expansion. For 75 per cent., the re-evaporation, if any, is so slight, as to leave a deficit of 12 per cent. by condensation, during expansion, compared with what was indicated as cut off. Now, the whole tenor of the evidence shows plainly, that the degree of condensation increases as the admission is shortened; and it may be safely inferred, that as 12 per cent. is shown to be the loss in full gear, there is at least 12 per cent. of loss for 50 per cent. of admission, cutting off at half-stroke. An approximate loss of 12 per cent. will, on this ground, be adopted for all admissions greater than half-stroke; and 12 per cent. will also be added to the indicated losses for shorter admissions, as an approximation to the real conditions.

INDICATED CONDENSATION OF STEAM IN OUTSIDE CYLINDERS, DURING THE ADMISSION OF THE STEAM.

Period of Admission in Parts of Stroke.	Total Volume by Expansion, the Initial Volume being = 1.	Indicated Condensation in Parts of the Indicated Steam cut off.	Approximate Proportion of Steam Condensed.	
			In Parts of the Indicated Steam consumed.	In Parts of the whole Steam consumed (including the Condensed Steam).
Per Cent.	Ratio.	Per Cent.	Per Cent.	Per Cent.
75	1·22	—12·0	12	11
65	1·40	— 5·2	12	11
50	1·54	0·0	12	11
40	1·78	9·4	21	17
30	2·07	19·9	32	24
20	2·45	34·1	46	32
12	3·17	61·1	73	42
1	2	3	4	5

The most direct test of the amount of loss from condensation of the steam during expansion, appears to be the mode adopted above, of comparing the water-equivalents, or the actual weights of the steam that is present in the cylinder, at the beginning and at the end of the expansion.

An exact conclusion as to the amount of condensation during expansion cannot be obtained from the loss of area in the indicator-diagram. The dotted line, E F G, added in Fig. 211, is not the curve that would actually have been described had there been no condensation, but such as might have been described by the quantity of steam which the final pressure, during expansion, proves to have been admitted. The loss by condensation could not have been less than shown by the shaded area, but was certainly greater in amount; for it appears that a portion of the steam admitted, and sometimes a considerable amount of it, is buried for ever, and is not resuscitated at all at the end of the expansion. This is proved by the great increase of back pressure that takes place when a high degree of expansion is used, from the lower temperature of the cylinder, as illustrated by the diagrams in Fig. 211, which must have been caused by the quantity of precipitated steam still remaining in the cylinder after expansion.

Column 4 contains the approximate losses as revised in the way above described, in parts of the indicated steam admitted. Adding the lost steam admitted to that indicated, the sum expresses the whole steam admitted and expended; and column 5 contains the percentage of approximate loss, expressed in terms of the whole steam so used, which is a more convenient form for reference. From this column it appears that, for 40 per cent. admission, 17 per cent., or one-sixth of the steam, is condensed; for 30 per cent., one-fourth; for 20 per cent., one-third; and for 12 per cent., or mid-gear, two-fifths, or not far from one-half.

It must be added that the foregoing deductions are based on steam-pressures under 60 lbs., generally about 50 lbs., during admission. For higher pressures, and admissions above half-stroke, the condensation is proportionally less, as will afterwards be shown.

Proof of the condensation of steam, in outside cylinders, is obtained by comparing the indicated consumption of steam with the measured consumption of water,—avoiding, of course, the error which may arise from the occasional priming of water from the boiler. For the purpose of testing this means of comparison, the following experiment was tried by the writer on the actual consumption of water by an outside-cylinder express engine, No. 42, for a trip of 105 miles, from Glasgow to Carlisle, on the Caledonian Railway, with a train averaging six and one-half carriages; the time of the trip being three hours twenty-two minutes, including five stoppages.

Indicator-diagrams were taken from the cylinder at intervals of one or two miles, and the notch of the expansion-gear observed for each diagram, and the points of the line where each change of notch was made.

The several points of cutting off, expansion, and compression, were accurately ascertained by means of the slow diagrams; from which were calculated the exact quantities and pressures of sensible steam actually consumed in each interval of the trip, and the water-equivalents for the several quantities of steam present in the cylinder; which, multiplied by the number of strokes of the two cylinders in each interval, gives the total quantity of water efficiently used as steam.

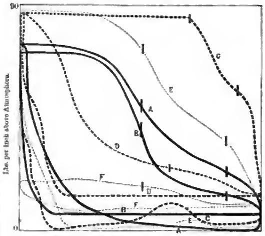
The following final results were thus obtained:—

		Average Admission in Parts of the Stroke.	Water used as Sensible Steam.	Water Consumed as Measured.	Errors.
		Per Cent.	Cubic Feet.	Cubic Feet.	
1	Glasgow to Motherwell, ... 16 miles; steam on, 14½ miles.	45	30·76	35·82	5·06 feet, or 14 per cent.
2	Motherwell to Carstairs, ... 15¾ miles; steam on, 14¾ miles.	54	43·91	48·85	4·94 „ 10 „
3	Carstairs to Beattock, ... 34 miles; steam on, 23½ miles.	50	57·28	67·74	10·46 „ 15½ „
4	Beattock to Carlisle, ... 39¼ miles; steam on, 38¾ miles.	40	62·42	79·50	17·06 „ 21½ „
	TOTAL. Glasgow to Carlisle, ... 105 miles; steam on, 91 miles.	45·6	194·37	231·91	37·54 feet, or 16¼ per cent.

The examination of the indicator-diagrams in the manner employed before, by comparing the initial and final water-equivalents of the steam during expansion, shows that at least 13 per cent. of this loss of 16¼ per cent. was due to condensation, and it is probable that no appreciable proportion was due to priming; indeed, the least loss was observed to take place with the least degree of expansion, and when the consumption of steam from the boiler was going on at the greatest rate, as is found on referring to the percentages of admission in the first column of the above table, which is the reverse of the effect that would have been observed if priming had been a material cause.

Experiments made by the writer with other outside-cylinder engines, or imperfectly-protected cylinders, corroborate the above deductions, obtained from the performance of No. 42; and they are still further corroborated by his experiments on inside well-protected cylinders, which show that, in ordinary good condition, there is no sensible excess of water of any importance actually consumed from the boiler, above what is estimated from the indicated steam passed through the cylinder. These results are also confirmed by the results of the trials of Sir D. Gooch, with the "Great Britain," and similar engines.

The increased back pressure of exhaust affords additional evidence of the presence of water in the cylinder. The back exhaust pressure is the consequence of the want of facilities for the timely discharge of the exhaust steam from the cylinder; and the impediments to its discharge are much increased by the presence of water amongst the steam, whether due to condensation or to priming. The presence of water is immediately made apparent by the increase in the back exhaust pressure, shown by the indicator-diagram, as the writer has on many occasions had an opportunity of observing. The effects of priming, from foulness of the water in the boiler, are shown in Fig. 215. A and B are indicator-diagrams taken from the well-protected cylinders of the "Orion," in which very little, if any, condensation could be detected. The



A and B, "Orion" Passenger Engine, . . 39 Miles per Hour.
C and D, No. 137, Goods Engine, . . 18 „
E and F, „ „ . . 15 „

Fig. 215.—Indicator-Diagrams to show increase of Back Pressure by Water in the Cylinder.

diagram A was taken before, and the diagram B after, the boiler was blown off and supplied with clean water, both being taken at the same speed, and showing 7 lbs. back pressure, caused by priming, in the former case.

The diagrams C and D, Fig. 215, show that the total quantity of water from condensation is considerably greater, with the greater degrees of expansion, where a smaller quantity of steam is admitted, and consequently the loss is more seriously felt. These diagrams were taken from the outside-cylinder goods engine, No. 127, working at the same speed up and down an incline on the Caledonian Railway; the diagram C, cutting off at two-thirds of the stroke, and the diagram D at one-sixth of the stroke. The latter, D, although it had the advantage of a much earlier exhaust, and only one-fourth of the quantity of steam to discharge, was affected with 10 lbs. more back pressure than the former, C, when working in full gear. This great back pressure was maintained over a continued run of twenty miles, when, of course, the cylinders had got into their working heat for that degree of expansion; and the inference is, that the steam was loaded with water of condensation (proved also by the expansion-curve), which was with difficulty expelled, and which only became proportionably less when the degree of expansion was diminished; and, consequently, the mass of steam increased that was to be cooled within the same superficies of cylinder.

That the total mass of the steam has much to do with the condensation is proved by the diagrams E and F, Fig. 215, taken under the same degree of expansion, and at the same speed, but with 75 lbs. and 20 lbs. steam, respectively, admitted to the cylinder. In the latter diagram, F, the back exhaust pressure is 7 lbs. greater than in the former diagram, E, although the total quantity of steam to be discharged was so much less. In the latter case, indeed, there was found to be an excess of 18 per cent. of the whole water used over the indicated steam expended, which was most probably altogether by condensation, as the rate of consumption was so moderate as to preclude any likelihood of priming.

Now, here is a case where, in the same class of engines, the back exhaust pressure increases as the quantity of steam to be discharged becomes less, notwithstanding that the facility for exhaust increases at the same time. This is clearly a case of water in the cylinder, the quantity of which increases with the degree of expansion; and the water is as clearly a precipitation of steam by condensation. Also, though a full admission of steam at higher pressures may reduce the proportion of condensation, yet, whenever expansive working is attempted by cutting off earlier, the heavy back pressure, and the course of the expansion-line, alike show that no pressure of steam, however high during the admission, can mitigate the evils of condensation in exposed cylinders.

Evidence from the Proportions of the Valve-gear.—The remarkable inversion, just discussed, of our ordinary experience with well-protected cylinders, whereby the back pressure rises with the degree of expansion, proves the necessity for more liberal proportions of valve-gear for outside cylinders than for insides, to afford a more free exhaust. The writer has invariably found that of the three fixed elements affecting the exhaust—namely, the sectional area of steam-port, the inside lead, and the area of blast-orifice (so long as the port is larger than the orifice)—it is the orifice alone which, in well-protected cylinders, rules the amount of exhaust back pressure; the wider the orifice the less the pressure, in the ratio of the fourth power of the diameter of the orifice, or the square of the area: whereas, in exposed cylinders, the back pressure is ruled both by the orifice and by the inside lead; the greater the orifice and the greater the inside lead, also, the less is the pressure. This is an important distinction, because it shows that, as inside lead is equal to the sum of the lap and the outside lead, and is, in fact, regulated by the lap, the lap of the valve is a very important element in the designing of outside cylinders, though practically a matter of indifference in insides. Accordingly, it has been found that in "Sharp's" inside-cylinder engines, on the Edinburgh and Glasgow Railway (now, 1870, a section of the North British Railway), which have only a ⅝th inch lap—probably the shortest lap in present practice for a 15 inch cylinder—the exhaust is as perfect as in the Caledonian passenger engines with 1¼ inch lap for the same cylinder. Further, in inside cylinders, with clean boilers, it is practically a matter of indifference what amount of wear or slugger may have taken place in the valve-gear, so far as concerns the exhaust. In outsides, on the contrary, it is a very important object to maintain the gearing in the highest order, so as to keep the inside lead, as the wear of the gearing directly reduces the lead, and thereby increases the back pressure. The Caledonian is perhaps the first line in this country on which the special advantage of long lap for outside cylinders was experienced. Nor need there be any apprehension of reducing the tractive power of an engine by increasing the lap, and thereby shortening the period of admission, because the same admission may be obtained by increasing the lead and the travel of valve in the same ratio with the lap; and it may be added, this may be simply done in existing link-motions, by extending the link beyond the eccentric-rod ends, and thereby increasing the range of the sliding blocks and the maximum travel.

The formidable degree of condensation which accompanies high expansion in partially-protected cylinders, accounts for the opinion held by men of experience, of the inutility, for economical objects, of cutting off the steam earlier than at half-stroke, since the advantage of expansive working, found with inside cylinders, is neutralized in outsides by the condensation. The remedy is to provide a complete and ample covering for the cylinders at the ends, as well as at the sides.

CHAPTER XX.

CONDITIONS OF STABILITY OF THE LOCOMOTIVE AS A CARRIAGE.

It is required of the locomotive, as a carriage or vehicle, that a sufficient weight should be carried by the driving wheels, to insure adhesion, and that the machine should run freely and steadily at all speeds.

Various circumstances operated to render the earlier classes of engines unsteady,—a short wheel-base, overhanging masses, and the unbalanced revolving and reciprocating masses of the crank, with the piston and its appendages. The evils were aggravated with outside, as compared with inside, cylinders, on account of the greater speed laterally of the swinging masses. These evils were sought to be remedied by various expedients,—the extension of the wheel-base, lowering the boiler, loading the foot-plate behind the fire box with cast iron, coupling the engine very stiffly to the tender, the use of three balanced cylinders, the stiffening of the frame, and the most prominent expedient of all—placing the driving axle behind the fire box. On this last system, the load was necessarily placed almost entirely upon the extreme axles, fore and hind, to insure a sufficiency of driving-weight; it followed that the horizontal leverage of the wheels with their loads, and their resistance to horizontal sinuous motion, were increased, in so far as a greater disturbing force is requisite to sway an engine about one of its extreme axles than about a central axle, and superior steadiness was thus obtained. This system, however, though it removed instability externally, left the swinging masses unbalanced; it also involved a longer wheel-base, and a heavier engine, than on ordinary plans.

The distinguishing features successively adopted in the construction of locomotive engines have already been fully described; but it may be well here to allude to them succinctly. The form of engine primitively adopted in the "Planet," by Stephenson, was made with four wheels and inside cylinders. This arrangement was succeeded, in Stephenson's practice, by an engine with a third pair of wheels behind the fire box, to check the vertical pendulous movement of the engine. Expansions of this normal arrangement were the inside-cylinder engines of Sharp, Wilson, Kitson, Bury, Stephenson, Hawthorn, and Gooch. The course of improvement and alteration in the arrangement of outside-cylinder engines was taken up by Stephenson, Trevithick, Allan, J. V. Gooch, and Crampton, and subsequently by many others. An adaptation of driving wheels behind the fire box to inside cylinders was made; and Stephenson's method of hind-drivers was applied, by Mr. W. B. Adams, to light tank engines on four wheels.

The remarkable uniformity with which the leading idea of a central driving axle in front of the fire box, initiated by Stephenson, is adopted by almost all the others, is noticeable. The reasons which led to this uniformity of practice are not difficult of appreciation, and they are worth some consideration, as bearing on the general question of single-wheel engines. In the first place, there is the demand for a sufficiency of driving-weight to supply the required adhesion for traction; but the apportioning of driving-weight to one pair of wheels must be executed with a regard to the requirements of the others, which are carrying wheels simply. The function of the front wheels is to lead the engine; and that of the hind wheels, as carriers, is mainly to steady it. Thus the front wheels require a greater load than the hind wheels, but not so much as the drivers; and the drivers stand first, the leaders next, and the trailers last in the order of loading.

It naturally follows that, for the driving wheels of single engines, the best situation is at some little distance behind the centre of gravity of the whole machine, because the supply of load to the driving wheels in that situation is the most direct; and, in the adjustment of the driving load, the loads on the other wheels are individually less affected than when the drivers are far from the centre of gravity.

It might be said, as a precautionary suggestion, with reference to the system of central driving wheels, that they should be placed some distance behind the centre of gravity, to secure sufficient weight on the leading wheels. This condition is easily met in practice, because, conveniently enough, the centre of gravity of ordinary engines is situated 3 to 4 feet in advance of the fire box, and there is room for the driving axle behind it.

Whereas, when the driving axle is behind the fire box, the centre of gravity lies midway between the extreme axles, and there is as much weight imposed—unnecessarily, but unavoidably—on the leading as on the driving wheels. Hence the first objection to the system introduced by Mr. Crampton—a deficiency of driving-weight, or the alternative—an extra weight of engine. Comparing that engine with those introduced by Mr. Bury—the gross weight of the former is 27 tons, of which 11½ tons rest on the front axle, 11½ tons on the driving axle, and 4 tons on the middle axle—the gross weight of Mr. Bury's engine is 20½ tons, of which there are 12 tons driving-weight, 6 tons leading, and 2½ tons trailing. Thus, with equal driving-weights, which ought to mean equal powers, Mr. Crampton's engine is 6½ tons heavier than Mr. Bury's.

In Mr. Crampton's engines, again, the load being thrown mainly on the extreme axles, the engine runs heavily and severely along curves, which is proved in practice by the greater wear of the leading-wheel flanges in them, than in ordinary engines.

Moreover, in this system an injurious extension of wheel-base is incurred, through the necessity for placing the leading wheels far forward, to relieve them as much as possible of what is, to them, superfluous load. The labour of passing curves also increases with the extension of the base.

All these objections are avoidable in the ordinary system with central driving wheels. The one primary and sufficient condition is, that the revolving and reciprocating masses of the pistons, piston-rods, cross-heads, connecting-rods, and cranks, should be balanced in the wheels. This condition was pointed out many years ago, by Mr. Stephenson, in his evidence before the Gauge Commissioners, in 1845; and he at the same time exposed the fallacy, that the action of the steam on the pistons had anything necessarily to do with unsteadiness. "When the steam presses upon the piston," he says, "it at the same time presses against the lid of the cylinder; the action and the reaction must be equal. Therefore, I do not believe that it is the steam that causes the irregular action, but I believe it to be the mere weight of the pistons themselves; and if we could contrive to balance the pistons by the weight upon the wheel, we should get rid of that very much." A complete balance can be effected; and it was done by the writer, for the first time in this country—so far as he is aware—in the earliest outside-cylinder engines of the Great North of Scotland Railway, which were designed by him. These engines run with absolute steadiness, at the highest speeds attained on that line, without the least internal disturbing action of any kind. They are further noticed in Chapter XXV.

All classes of engines, with inside or outside cylinders, single or coupled wheels, may be satisfactorily balanced on the principle above indicated.

Experimental Evidence.—As the wear of the leading wheels of engines may be accepted as an index to the labour of running, the writer produces a comparative statement, made in 1856, of the working of three classes of express engines on the North British Railway, showing the usual number of miles run by the leading wheels of each class, until they required re-turning; also the nature of the wear of these wheels.

The leading wheels of ordinary express engines, with inside cylinders, unbalanced, run 8,500 miles; they were turned up, on account of the flanges wearing thin, in nine cases out of twelve.

Those of the "Queen" express engine, with inside cylinders, unbalanced, ran 7,000 miles; they were turned up, on account of the flanges wearing thin.

Those of Mr. Crampton's engine ran 6,000 miles, and were also turned up, on account of the flanges wearing thin.

Thus, the "Queen" engine ran its leading wheels 17 per cent. more miles than Mr. Crampton's; and the ordinary express engines 42 per cent. more miles,—notwithstanding that Mr. Crampton's engine ran much more steadily than the other two. The cause of the inferior duration of Mr. Crampton's leading wheel-tyres is to be found in the extra labour of running round curves.

Economy of fuel is materially promoted by the correct equilibration of engines. With Mr. Beattie's permission, the writer, in 1856, balanced the "Canute" engine, on the London and South-Western Railway. This engine had previously a balance-weight of 85 lbs. applied within the rims of the driving wheels. New weights were put in, weighing 186 lbs. for each wheel, and balancing the whole mass acting at the crank pin. The engine ran so much more steadily and freely with the new balance-weights, as to take the engineman by surprise. On the first day after the alteration, it considerably overshot the stopping stations, although the steam was shut off and the brakes applied at the usual distances from the stations.

In working the main-line trains, the average performance of the "Canute," taken from the Company's books, was as follows:—For four months, ending January 30, 1856, she took an average train of twelve carriages, with 20·9 lbs. of coal per mile. Early in February, the balance-weights were inserted, and subsequently, up to March 26, 1856, she took an average train of 10·6 carriages, with 17·9 lbs. of coal per mile. The subsequent average train was thus less by 1·4 carriages, and the fuel was less by 3 lbs. per mile. Making an allowance of ·15 lb. of coal per ton, which, at 6¼ tons per carriage, would be 1·3 lb. per mile, there remains an economy of 1·7 lb. of coal per mile due to the extra balance-weights. But this does not adequately represent the economy, because the weather subsequently was much more severe than previously to the equilibration. Accordingly, the total average performances of three other main-line passenger engines, during the same periods, which were in the same condition as to equilibration throughout, show that the average trains fell from eleven to ten carriages, and the consumption rose from 19·3 lbs. to 20·8 lbs. of coal per mile. Allowing for the difference of trains as before, the equivalent consumption in the second period would be 21·8 lbs., and the increase would then be 2·5 lbs. per mile, representing the increased severity of the weather. Adding this to 1·7 lb., the saving already ascertained, the sum, 4·2 lbs. per mile, represents the presumable saving in fuel due to the complete equilibration of the "Canute," amounting to 20 per cent. of the original consumption.

The "Norman," an outside-cylinder four-coupled goods engine, on the South-Western Railway, was about the same time equilibrated according to the plans of the author. Such engines, unbalanced, are the most unstable of all, and the saving by equilibration should be all the greater. The engine and tender were taken alone on a trial run, with the balance-weights complete, and they ran at speeds of more than sixty miles per hour, with perfect steadiness, excepting the disturbances due to the road, there not being the least oscillating motion of any kind. The counter-weights were then taken out of the wheels, and the engine and tender again ran out alone; but so violent was the oscillation of the engine, both laterally and fore-and-aft, and so violent also the concussions between the engine and tender, that the engineman could not venture to exceed a speed of about twenty miles per hour; and two strong hooks between the engine and tender were successively broken across, owing to the lurches of the engine.

Three other goods engines of the class of the "Norman" were equilibrated similarly to the "Norman," and the following tabular statement, showing the results of the performance, indicates a decided economy of fuel by correct equilibration, as compared with the want of it:—

Name of Engine.	Before Equilibration.			After Equilibration.		
	Miles Run.	Average Load.	Coke and Coal per Mile.	Miles Run.	Average Load.	Coke and Coal per Mile.
	Miles.	Waggons.	Lbs.	Miles.	Waggons.	Lbs.
Norman, .	26,888	31·2	26·1	21,319	27·5	23·4
Saxon, . .	36,418	28·0	28·3	12,279	25·9	23·6
Albion, . .	19,042	26·4	28·4	13,662	27·3	20·9
Samson, .	26,143	26·4	24·1	12,434	27·7	23·9
Totals and Averages,	108,491	28·0	26·7	59·694	27·1	22·95

The averages of these four goods engines show that, before equilibration, the consumption of fuel was nearly 27 lbs. por mile, with an average train of twenty-eight waggons; and after equilibration, 23 lbs. per mile, with an average train of twenty-seven waggons. Allowing 1 lb. of fuel extra for the same number of waggons, there remains a saving of 3 lbs. per mile, or 11 per cent.

The beneficial results of connecting equilibrating locomotive engines have been fully demonstrated by Le Chatelier, who found that an engine, unbalanced, could not attain to so high a speed as when balanced. He worked a six-coupled goods engine with outside inclined cylinders, and 4 feet 3 inch wheels. This class of engine had been found very unsteady on the rails, and had required frequent repairs. Counter-weights, equivalent to 1,100 lbs. at the crank-pin, were equally distributed between the three wheels on each side of the engine—well worn by long service. They were placed exactly opposite the cranks, the total weight being less than would exactly have balanced the engine. With a train of forty-four waggons, at thirty miles per hour, the engine ran with steadiness, though the wheels had already been well worn by long service, and the axle boxes had considerable play. The counter-weights having then been removed, the engine was again set to work with the same train, when it was found impossible to attain a higher speed than twenty-five miles per hour, as the engine ran so unsteadily, and was affected with violent oscillation and fore-and-aft movement. The economy of fuel effected by equilibration is testified by the results of performance of this engine before and after having been equilibrated: thus,—

January, 1849.—50·3 lbs. coke per mile, without counter weights.
February, 1849.—42·3 lbs. coke per mile, mean of twelve trips, of which ten only were made with counter-weights.

CHAPTER XXI.

INTERNAL DISTURBING FORCES IN A LOCOMOTIVE ENGINE IN MOTION—RULES FOR EQUILIBRATION.

The operation, referred to in the preceding chapter, of the reciprocating parts of the mechanism as disturbing causes, tending to excite angular or pendulous movements horizontally and vertically, and also longitudinal fore-and-aft movements, is readily explained. The reciprocating parts are, during each stroke, started from a state of rest with accelerated motion, which attains to its maximum at half-stroke; during the second half of the stroke they are retarded in motion, and finally brought to a state of rest at the end of the stroke. Such proportion of the pressure of the steam in the cylinders as is necessary to impress upon the pistons and other reciprocating parts the accelerated movement, is necessarily exerted upon the cylinder-cover—acting as a spring as between the cover and the reciprocating parts—and through the cylinder upon the body of the engine. The whole machine, therefore, supposing there is no counterbalance-weight, is swayed to one side. During the second half of the stroke, on the contrary, the momentum acquired by the reciprocating parts is delivered to the crank-pin and axle, and thus to the whole machine, by which it is caused to sway horizontally in the contrary direction. This double action is repeated during the return-stroke of the piston, and thus it follows that the engine is subjected to four distinct and contrary swerving impulses during one revolution of each crank, or to eight impulses in all, reckoning for two cylinders, during one revolution of the driving wheel.

When the engine runs without steam, the circumstances under which the accelerated movement of the pistons is acquired are different from those which exist when the engine is running with steam on the pistons. In the absence of steam on the pistons, the labour of starting and accelerating the motion of the piston devolves upon the crank on the driving axle; so that the swaying strain is transferred from the end of the machine, where the cylinder is placed, to the middle. This change of circumstances, though it does not affect the longitudinal reciprocating movement, increases the lateral oscillation, because the disturbing force has, in the latter case, more power over the mass of the machine.

With respect to disturbance arising from the action of the steam on the pistons, apart from that which is absorbed in accelerating them, the angularity of the connecting-rod gives rise to an upward pressure upon the guide-bars, when the engine is going ahead, tending to lift the engine off its leading springs at half-stroke. The alternate heaving and sinking so caused is called the *pitching* of the engine; and further, as the upward movement is not simultaneous for the two cylinders, but successive, the heaving of the engine takes place alternately on the two sides, and causes a *rocking*, or lateral rolling motion, on the springs.

There is another source of unsteadiness in the unequal quantities of steam admitted to the two ends of the cylinder with imperfectly-arranged valve-gear, by which a greater quantity of steam is admitted into the cylinder, and a greater mean pressure is reached, during the front-stroke of the piston, than during the back-stroke. As the two front-strokes of the two pistons take place consecutively in the course of a revolution of the driving wheel, and alternately with the two back-strokes of the two pistons, it follows that a varying impulsive force must be exerted upon the engine, resulting in a longitudinal reciprocating movement of the whole machine when in motion.

Practically, it is found that for engines with inside cylinders, a counter-weight in the wheels, equivalent to three-fourths of the gross disturbing weight on each side of the engine, is sufficient to insure the external stability of the engine on the rails. For engines with outside cylinders, the counter-weights should be equivalent to the whole disturbing weights; or, at least, for single engines, not less than seven-eighths of the weight. The balance-weight should be distributed over at least two or three spaces, to distribute and reduce the unequal wear of the tyres by vertical action, and the tendency to slip at high speed. The more nearly the width apart of the cylinders is equal to that of the wheels, the more nearly exactly may both the longitudinal and lateral actions be balanced by counter-weights in the wheels. Thus it is that outside cylinders are susceptible of a better balance than insides, and the closer that inside cylinders are placed to each other, the less perfectly can they be balanced in the wheels. It follows, too, that for inside cylinders, the best position of the counter-weight on each side is not diametrically opposite the near crank, but at an angle of divergence from the exactly opposite radius, and diverging also from the far crank; in order that, in fact, the counter-weight on each side may not only, for the most part, balance the near crank and its connections, but also in part the far crank. The annexed diagram, Fig. 216, illustrates the relative positions of counter-weights in the wheels for inside cylinders, 30 inches apart centres, and wheels 55 inches apart centres, on the 4 feet 8½ inch gauge. Let A, B, be the right and left-hand cranks in side elevation, respectively in a horizontal and a vertical position; then the

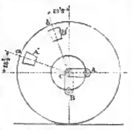


Fig. 216.—Inside Cylinders—Relative positions of Counter-weights in Wheels.

counter-weight A′, for the right-hand wheel, lies in the direction C *a*, at 18½° from the centre line C A, diverging from the crank B; and the counter-weight B′, for the left-hand wheel, lies in the line C *b*, 18½° from the centre line C B. Thus, the two counter-weights on the opposite sides of the engine incline towards each other, when seen in side elevation, and their directions, C *a*, C *b*, form an angle of 53°, or less than a right angle, by as much as twice 18½° or 37°.*

The following practical rules for equilibrating locomotive engines are derived from *Railway Machinery*:—

RULE I.—*To find the Counter-weight for Outside-cylinder Single Engines.* Find the total weight, in lbs., of the revolving and reciprocating masses, for one side—namely, the piston and appendages, connecting-rod, crank, and crank-pin—multiply by the length of crank in inches, divide by the radical distance, in inches, of the centre of gravity of the space to be occupied by the counter-weight. The result is the counter-weight, in lbs., to be placed exactly opposite to the crank.

Note 1.—The "crank," for the purpose of calculation, is to be taken as the extra material applied between the spokes of the wheel to form the crank; and the weight and centre of gravity of this extra material having been ascertained, the calculated equivalent-weight, as applied at the crank-pin, is to be employed in the calculation.

RULE II.—*To find the Counter-weights for Outside-cylinder Coupled Engines.* Find the separate revolving-weights, in lbs., of crank-pin, coupling-rods, and connecting-rods, for each wheel; also the reciprocating-weight of the piston and appendages, and half the connecting-rod; divide the reciprocating-weight equally between the coupled wheels, and add the aliquot part, so allotted, to the revolving-weight on each wheel. The sums so obtained are the weights to be balanced at the several wheels, for which the necessary counter-weight may be found by RULE I.

RULE III.—*To find the Counter-weight for Inside-cylinder Single Engines.* 1st. To find its value. Find the total weight, in lbs., to be balanced on each side, as in RULE I.; multiply it by the sum of the widths apart of the centres of the cylinders, and of the wheels, in inches, and divide by twice the width apart of the wheels; subtract the quotient (A) from the total weight, leaving a remainder (B); square the quantities A and B, add the squares, and find the square root of the sum. This root is the resulting weight, in lbs., to be balanced at the crank-pin, for which the counter-weight may be found by RULE I.

2nd. To find its direction. Divide the greater weight (A) by the less (B). The quotient is the denominator of the fraction, of which the numerator is 1, which expresses the inclination of the direction sought, with the centre line of near crank, diverging from the off crank.

RULE IV.—*To find the Counter-weights for Inside-cylinder Coupled Engines.* Find the value and direction of the counter-weights for the inside revolving and reciprocating masses to be balanced, as in RULE III., and key the driving wheels on the axle in such positions as to place the outside cranks in the direction so found, or key on the cranks themselves, as required, if independent of the wheels; find the total weight of the outside cranks and coupling-rods, reduced, according to the note to RULE I., to the radius of the inside crank-pin; and, if less than the inside weight, subtract the outside weight from it, and distribute the difference between the coupled wheels, to be balanced according to RULE I.; or, if greater, balance the difference by counter-weights opposed to the outside cranks.

Note 2.—The counter-weight for inside cylinders may be found approximately by assuming three-fourths of the whole disturbing weight as the weight to be balanced in the wheel.

3. Inside-cylinder coupled-engines, as they stand, usually fall within the requirements of Note 2.

4. Though Note 2 contains a good general rule for stability, the other rules should be employed where exact equilibrium is required, so as to balance as well as possible every internal strain.

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* For a full analysis of the principles of equilibration, as applied to locomotive engines, see the writer's work on *Railway Machinery*, page 165.

CHAPTER XXII.

EXAMPLES OF EXPRESS LOCOMOTIVE ENGINES WITH SINGLE WHEELS.

The express engines selected for illustration were designed and constructed by the following engineers for the annexed Railway Companies:—

1. Express Passenger-locomotive, by Mr. John Ramsbottom, Locomotive Superintendent, London and North-Western Railway. Plates I. and II.

2. Express Passenger-locomotive, by Messrs. Beyer, Peacock, & Co., Manchester, for the South-Eastern Railway of Portugal. Plates III. and IV.

3. Express Passenger-locomotive, by Mr. Benjamin Conner, Locomotive Superintendent, Caledonian Railway; constructed by Messrs. Neilson & Co., Glasgow. Plates VII., VIII., and XIV.

I.—Express Passenger-locomotive, by Mr. John Ramsbottom, for the London and North-Western Railway. Plates I. and II.

Area of Grate, ...	14·9 square feet.
Heating Surface, ...	1,098 "
Diameter of Cylinders, ...	16 inches.
Stroke of Pistons, ...	24 "
Diameter of Driving-wheels, . . .	7 feet 7½ inches.
Length of Wheel-base, ...	15 " 6 "
Total Weight of Engine in working order,	27 tons.
Do. Tender do. do.,	17½ "
Do. Engine and Tender do.,	44½ "
Weight on Leading-wheels of Engine, .	9 tons 8 cwts.
Do. Driving-wheels, ...	11 " 10 "
Do. Trailing-wheels, . .	6 " 2 "
	27 tons 0 cwts.
Weight of Engine, empty, ...	24 10 "

This engine, the "Lady of the Lake," has already been partly described at pages 135 and 136. It is one of a numerous class designed with special regard to the running of express trains on the northern division of the railway. It may be favourably contrasted with another express engine, No. 373, both engines having been exhibited by the London and North-Western Railway Company in 1862. No. 373 was designed with special regard to the running of the express trains on the southern division. Only three engines were made of this class, as it was found to be heavy on the road and extravagant in fuel. It is an engine of maximum dimensions, and, indeed, beyond the capacity of the narrow gauge for proper working. With cylinders of 18 inches diameter, and stroke of 24 inches, 7 feet 6 inch driving-wheels, and 26 square feet of grate, it weighs 34¾ tons in working order, of which there are fully 14½ tons of driving-weight at one pair of wheels. With the tender, weighing 25 tons with fuel and water, the total mass to be moved amounts to about 60 tons, exclusive of train-weight. The boiler was designed for burning coal. The combustion-chamber having been found insufficient for the purpose of mingling the combustible gases and consuming them before entering the flue-tubes—which, indeed, was scarcely to be expected, the chamber being only 2 feet 8 inches in length—the fire-box was fitted with fire-brick arches inside, and with deflecting plates in the doorways. The grate was 7 feet long, in two parallel strips, and the enormous amount of heating surface, 242½ square feet, has thus been attained in the fire-box and the combustion-chamber, making what is called "direct" heating surface,—more than has before been attained on the narrow gauge. But the advantage of "direct" surface, or that which is exposed to the radiant heat of the fire, depends entirely upon its being within a reasonable distance from the fuel on the grate; whereas, in this fire-box, the crown is 6 feet 3 inches above the grate; and the upper portions of the surface are nearly inoperative for evaporation. The first concern is to bring the surface near to the fire, or the fire near to the surface. It is thus that the efficient performance of Mr. Cudworth's boiler (Plate XXVIII.), with the elevated inclined grate, is to be accounted for, since in that boiler, not only the lateral but the horizontal surface is thoroughly heated by direct action. The great extension of fire-box and combustion-chamber in No. 373 led to the curtailment of the flue-tubes to 9 feet 4 inches in length, and to the attempt to compensate for this by packing 214 of them together at ¼ inch distances apart; making 980 square feet of heating surface. There is a general belief that heating surface is mechanically the equivalent of evaporating power. This is a mistake. Besides surface, circulation is wanted,—the circulation of the water to, and the steam from amongst, the tubes. The evaporative power of a less number of tubes, placed at ¾ inch apart, would have been decidedly greater than that of the tubes as they are. The driving and leading springs are connected by a compensating beam, which must have given an unsafe freedom of action to the engine at high speed.

Contrasting the "Lady of the Lake" with No. 373, its weight in the first place is only 27 tons in working order, and the tender 17½ tons, making a total of 44½ tons, as against 60 tons for No. 373. The fire-box is of the ordinary form, with little more than half the grate surface of No. 373, and is fitted for coal burning, with a brick arch and two square air-openings in front. The heating surface of the fire-box is little more than one-third that of No. 373; and there are fewer tubes, but then they are ⅝ inch apart. No. 373 gains the advantage of the "Lady" in the greater size of blast-orifice, which is 5½ inches diameter for the former, and 4½ inches for the latter, owing to the larger grate of No. 373, which does not require the same sharpness of blast to draw the air through. The proportions in question are here brought together for comparison:—

	No. 373.	Lady of the Lake.
Area of Fire-grate, . . .	26 sq. ft.	15 sq. ft.
Heating Surface of Fire-box, .	242·5 "	85 "
Do. Tubes, .	980·5 "	1,013 "
Do. Total,	1,223·0 "	1,098 "
Driving-weight, . . .	14 tons.	11¼ tons.
Diameter of Blast-orifice, . .	5¼ inches.	4¼ inches.

The annexed table contains the results of the performance of engines of the same classes respectively, in which the weights of the trains are averaged from actual observation, being 8½ tons per vehicle, with passengers and luggage; the vehicles averaging 6 tons 11 cwts. nett, and 39 cwts. per vehicle being allowed for the carried weight. It appears that whilst the Southern engines have only 36 per cent. more train-load, they consume 55 per cent. more coal than the Northern engines. The consumption per ton of gross weight, including the engine and tender, is at the rate of 0·26 lb. per ton per mile for the Southern, and 0·23 lb. per ton per mile for the Northern,—which shows pretty clearly that there is no mechanical superiority in the Southern boiler, with its extensive direct heating surface, over the Northern boiler of the ordinary type.

RESULTS OF THE PERFORMANCE OF ENGINES OF THE CLASSES OF "NO. 373," AND THE "LADY OF THE LAKE," EXPRESS ENGINES, ON THE LONDON AND NORTH-WESTERN RAILWAY, IN 1862–63.

Engine.	Months at Work.	Miles Run, to 31st October, 1862.	Average Weight of Train (at 8½ Tons per Vehicle).	Gross Weight of Engine, Tender, and Train.	Consumption of Coal.		
					Per Mile.	Per Ton of Train per Mile.	Per Ton Gross of Engine, Tender, and Train per Mile.
Southern Division.	Months.	Miles.	Tons.	Tons.	Lbs.	Lbs.	Lbs.
No. 972, ...	6	10,560	95·07	...	42·13	0·44	...
No. 975, ...	9	23,906	101·55	...	41·48	0·41	...
Mean of Engines of the class of "No. 373,"			99·56 (Averaging 11·7 Vehicles.)	193·36	41·68	0·42	0·26
Northern Division.							
No. 44, "Harlequin,"	32	102,714	72·25	...	26·85	0·37	...
No. 196, "Leander,"	22	76,908	72·35	...	25·70	0·36	...
No. 229, "Watt,"	34	105,717	74·00	...	27·43	0·37	...
Mean of Engines of the class of the "Lady of the Lake," .			72·93 (Averaging 8·58 Vehicles.)	117·35	26·77	0·37	0·23

Regarding, again, the engines as carriages, the great height of the centre of the boiler of No. 373—7 feet 5¼ inches above the level of the rails—is excessive, and tells upon the rails when the engine sways. In the "Lady of the Lake," though she has driving-wheels as large as those of No. 373, the centre is 11 inches lower; and thus, in conjunction with a compact wheel-base and a balanced driving-wheel, produces a safe, steady, and easy-running engine.

The "Lady of the Lake" is fed by two of Giffard's injectors. It is fitted with Mr. Ramsbottom's duplex direct-action safety-valve, in which a pair of valves are pressed down by a cross-bar and a helical spring attached to the middle of the bar, between the valves,—a decidedly superior system to the weighted lever. It cannot be tampered with, and is much more prompt in discharging an excess of steam, as it opens wider for a given excess of pressure. It is similar in principle to Mr. J. V. Gooch's single direct-action valve. The smoke-box is fitted with a descending hopper, through which ashes and ignited cinders are allowed to escape from the smoke-box; thus, overheating is prevented. The cylinders and valve-faces are lubricated by means of gravity-lubricators, in which the water held in suspension in the steam is deposited in the vessel containing the oil, displacing it, and forcing it over into the cylinder, whence it is swept off by the steam over the working surfaces. The pistons are of cast-iron, in one piece, fitted with Mr. Ramsbottom's packing-ring, made of square wrought-iron wire, hard-drawn, and bent to such a form as to give uniform pressure round the piston. The reversing gear is worked by a screw and hand-wheel, instead of the usual long lever and notched sector, than which it is much more easily worked, saving the engineman a great deal of fatigue. This reversing gear is applied to nearly 200 engines.

The tender of the engine, exhibited with it, had six wheels, and weighed, empty, 9¼ tons; full, 17½ tons—the load equally distributed on the wheels. It is fitted with Mr. Ramsbottom's apparatus for picking up water whilst running—illustrated at page 101—in which a scoop is let down from the bottom of the tender, and dips into the water, contained in a long open trough between the rails, from which it is scooped up into the tank while running. A minimum speed of twenty-two miles per hour is required for this operation. Three such apparatus have been laid down, and are at work, on the London and North-Western

Railway; and their advantages are, that the size and weight of the tender, for running a given distance, may be reduced, the number of stoppages lessened, and time saved. An express engine has thus been enabled to run the whole distance from Holyhead to London—264 miles —in one continuous run, at an average speed of forty-two miles per hour.

II.—Express Passenger-locomotive, by Messrs. Beyer, Peacock, & Co., Manchester, for the South-Eastern Railway of Portugal. Plates III. and IV.

Area of Grate, ...	18 square feet.
Heating Surface, ...	1,339 „
Diameter of Cylinders, ...	16 inches.
Stroke of Piston, ...	22 „
Diameter of Driving-wheels, . .	7 feet.
Length of Wheel-base, . . .	15 feet 4 inches.
Weight on Leading-wheels, . .	9 tons 4 cwts.
Do. Driving-wheels, . . .	11 „ 12 „
Do. Trailing-wheels, . . .	7 „ 10 „
Total Weight of Engine in working order,	28 tons 6 cwts.
Total Weight of Engine, empty, . .	26 „ 13 „

This engine is a type of the prevalent style of English inside-cylinder express engines, with single wheels. The framing is composed of two pairs of longitudinal bars or slabs, running straight from end to end, cross-braced by the cylinders, the foot-plate, and various cross-plates. The broad and comprehensive slab frame-plate here noticeable was introduced by Mr. Beyer many years ago, and is now generally adopted in English practice. The extension of the two inside longitudinal slabs alongside the fire-box, formerly stopping short in front of it, is an important concession to good principle, and is a first-rate practice, as it connects the steam-cylinders, driving-axle, draw-plates, and bolts directly and immovably together, and bears and transmits the entire strain of the steam in the cylinders, and the tractive force, to the train. The boiler is thus relieved of all strain from the working parts, from which formerly it was not free, and it suffered accordingly. The driving-axle is made with only two bearings, inside the wheels, for which the guards are forged on the inside frame-plate, and the fore and hind axles have their bearings outside the wheels—an arrangement originated by the late John Gray, and now generally adopted for its simplicity, and for the greater firmness of the frame, and the increased duration or "life" of the crank-axle. The steam-strain is, in fact, confined to the two inside frame-plates, and to the inside bearings of the axle, close to the cranks. The fore and hind wheels are 3 feet 9 inches diameter, the drivers being 7 feet. The fore-wheels have been thought rather too small, and no doubt there is space in the engine for enlarging them; but they would have departed from uniformity with the six tender-wheels, which also are 3 feet 9 inches. The least that can be said is, they run fast enough.

The short cast-iron blast-pipe, reaching just above the level of the upper row of the flue-tubes, is to be remarked. This level of blast-orifice gives the best results, creating a better draft with a wider orifice, as compared with higher blast-pipes, and was arrived at by Mr. Peacock by means of a series of well-arranged experiments on the Manchester, Sheffield, and Lincolnshire Railway. The low blast-pipe is now generally employed. The first blast-pipes were carried some distance into the chimney, and had very contracted orifices and a sharp blast, causing much back pressure on the pistons.*

The "straight-link" motion of Mr. Allan is applied to work the slide-valves of this engine. It is a pleasing variety of the expansion-link, and has been described in another place.

The boiler, facilitated by the width of gauge—5 feet 6 inches—has a large square fire-box, 4 feet 10 inches each way, over the outside shell, and has a large diameter of barrel, 4 feet 2 inches, which gives abundance of steam-room and water-space. Nevertheless, for effectiveness of heating surface, a fire-box of oblong form would have been better. There are 219 2-inch flue-tubes, placed at $\frac{7}{16}$ths of an inch apart. Had the tubes been only 1¾ inch diameter, which, upon the whole, is the best size, and placed in the same position, the extra clearance so obtained for circulation would have improved the evaporative efficiency. The fire-box is adapted for coal-burning, with a brick arch, a deflector-plate from the doorway, and a sliding door, on the plan first used on the Midland Railway.

III.—Express Passenger-locomotive, by Mr. Benjamin Conner, Locomotive Superintendent, Caledonian Railway, constructed by Messrs. Neilson & Co., Glasgow. Plates VII., VIII., and XIV.

Area of Grate, ...	13·9 square feet.
Heating Surface, ...	1,172 „
Diameter of Cylinders, ...	17¼ inches.
Stroke of Pistons, ...	24 „
Diameter of Driving-wheels, . .	8 feet.
Length of Wheel-base, ...	15 feet 8 inches.
Total Weight of Engine in working order,	30 tons 13 cwts.
Weight on Leading-wheels, . . .	9 tons 5 cwts.
Do. Driving-wheels, . . .	14 „ 11 „
Do. Trailing-wheels, . . .	6 „ 15 „
	30 tons 13 cwts.
Weight of Engine, empty, . . .	27 „ 5 „

The class of engines of which this is one, were designed with a view to economy rather than speed, and are employed on main-line service between Glasgow and Carlisle, taking their turn of all the trains, fast and ordinary. The experience that had previously been acquired of the durability and general economy of the 7-feet driving-wheel over the original 6-feet wheel passenger engines of the line, suggested to the designer that a further enlargement of the driving-wheel would be advantageous and satisfactory. It is a fine, well-constructed engine, standing gracefully on its wheels, large yet compact, and qualified to run at any speed with ease and steadiness. Its striking appearance, with its magnificent wheel—the largest in the Annex of the Exhibition of 1862—secured for it the choice of his late Highness the Viceroy of Egypt, who wanted an express engine to carry him at seventy miles per hour. This engine has grown out of the old Crewe pattern of engine, originally introduced on the Caledonian line by the late Mr. Locke, the chief engineer, and successively modified by Mr. Robert Sinclair, and by Mr. Conner, to meet the growing requirements of the traffic. This engine weighs, in working order, 30 tons 13 cwts., of which the driving-weight is in excess of any other in the building, if that of the North-Western inside-cylinder engine be excepted. The largeness of the driving-wheel, 8 feet 2 inches, may prevent in some degree the injurious effects of concentrated loads on the permanent way.

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* For a detailed account of Mr. Peacock's experiments, see *Railway Machinery*, page 134.

The counter-weights are compactly forged into the rims of the wheels, extending over a third of the circumference; and though a little more weight is thus requisite to complete the balance, it sweeps gently over the rails, when the engine is in motion, without the sledge-hammer effect of a revolving cubical mass, compressed into the space between two or three spokes. For such a large wheel, also, the constructors have, with commendable caution, thickly planted the spokes, at 10-inch centres, on the rim: the stiffness of a spoke decreases in a rapid proportion with its length,—inversely as the cube of the length,—so that a spoke for an 8-feet wheel will be only half as stiff as a spoke of the same scantling for a 7-feet wheel. The driving-axle is of cast-steel, manufactured at the works of the Caledonian Railway; the tires are of Krupp's steel.

The framing of the old Crewe engine is retained in this one, giving outside bearings to the fore and hind axles, and insides to the driver. In this respect the "Lady of the Lake," a recent Crewe engine, is at variance with the older practice, as it has but one longitudinal frame-plate on each side, with inside bearings for all the wheels; and, with outside cylinders, it may very properly become a question whether the cylinders should be, as they are in the Caledonian engine, so rigidly united to the outside frame-plates, which carry two pairs of wheels and axles, and which, of course, transmit the shocks of the road to the cylinders. The foot-plate is judiciously provided with a housing for the engineman and stoker—a most important provision for their comfort, and for the safety of railway trains. Plain fence-plates are now commonly applied to engines across the back of the fire-box, and they are useful. But the housing is better, and it is marvellous that the application of so needful a protection should have been so long neglected.

The slide-valves are made with 1½ inch lap at each end, in pursuance of the conclusion arrived at by Mr. Sinclair, as to the peculiar importance of long lap and long travel for the valves of outside cylinders, unquestionably facilitating the free exhaust of the steam, at high speed, from exposed cylinders, in which the steam is found to be more or less condensed. Moist steam is not so active as dry steam; and if not freely discharged, it exerts an excessive back pressure on the piston.

The boiler is fitted with the brick arch and door-deflector, already described, applied by Mr. Conner to all or most of his engines for burning coal; 192 flue-tubes, 1⅝ inch diameter, and placed at ⅝ inch clearances, in a 3 feet 10 inch barrel. Giffard's injectors are employed.

The average work of engines of this class on the Caledonian Railway is a load of nine heavy carriages, being composites of four compartments, and third class with five compartments. The running speed with local trains is about thirty-five miles per hour, and with fast trains forty miles per hour. The average consumption of coal in February, 1863, with prevalent high winds, is stated to have been 25 lbs. per train mile; in the summer of 1862 it averaged 21½ lbs. per mile. They frequently take without assistance fourteen loaded carriages up the Beattock incline, ten miles long, at the rate of thirty miles per hour. The gradient is 1 in 84 for four miles, and 1 in 75 for six miles.

CHAPTER XXIII.

EXAMPLES OF EXPRESS LOCOMOTIVE ENGINES WITH FOUR-COUPLED WHEELS.

Coupled express engines are probably more frequently to be met with now, than single-wheel express engines. They have come into use in consequence of the increasing weight and speed of express trains,—demanding the greater degree of certainty and power of adhesion obtained by coupling the wheels, as compared with single wheels.

1. Express Passenger-locomotive, by Mr. Robert Sinclair, for the Great Eastern Railway. Plates IX. and X.

2. Express Passenger-locomotive, by Mr. Joseph Beattie, Locomotive Superintendent, London and South-Western Railway. Plates XVI. and XVII.

3. Express Passenger-locomotive, by Messrs. Sharp, Stewart, & Co., Manchester. Plate XXIV.

I.—Express Passenger-locomotive, by Mr. Robert Sinclair, for the Great Eastern Railway. Plates IX. and X.

Area of Grate, ...	15·5 square feet.
Diameter of Cylinders, ...	16 inches.
Stroke of Piston, ...	24 "
Diameter of Driving-wheels, . . .	7 feet 1 inch.
Length of Wheel-base, ...	15 feet.
Total Weight, in working order, . .	30 tons.

In general design and arrangement, this engine is similar to the first engines designed by Mr. Sinclair for the Caledonian Railway, after the original Crewe pattern of engine:—with double longitudinal frame-plates on each side, between which the cylinders are secured; a driving-axle well forward, and box-link motion. Mr. Sinclair early adopted the box-link, and has uniformly adhered to it. The fire-box is fitted with a simple smoke-baffling door, on Frodsham's system, with a valve adjusted at such an angle as to direct the air entering by the doorway downwards upon the fuel. In the absence of a brick arch, or any other means of controlling the course of the smoke and air which meet in the fire-box, this expedient is not sufficient for the purpose.

II.—Express Passenger-locomotive, by Mr. Joseph Beattie, for the London and South-Western Railway. Plates XVI. and XVII.

Area of Grate, ...	16·1 square feet.
Heating Surface, ...	1,149 "
Diameter of Cylinders, ...	17 inches.
Stroke of Pistons, ...	22 "
Diameter of Driving-wheels, . . .	7 feet.
Length of Wheel-base, ...	14 feet 3 inches.
Total Weight, in working order, . .	33 tons.

The chief features of this engine are comprised in the boiler, the principle of which has already been described (page 136). The barrel is 3 feet 9 inches in diameter; and there are 248 flue-tubes $1\frac{3}{16}$ inches in diameter, with $\frac{3}{16}$ inch clear space between the tubes.

Mr. Beattie's name has been long and favourably known as that of a leader in the economies of locomotive power. In 1856, the writer was enabled to make a series of trials of one of Mr. Beattie's coal-burning passenger-locomotives—the "Canute," on the London and South-Western Railway—the results of which may be recapitulated here. The fire-box of that engine is differently arranged in some details from that which is illustrated in this work. It is large, and is divided transversely into two compartments by a mid-feather. There is a combustion-chamber in the barrel of the boiler, terminating in flue-tubes which proceed to the smoke-box. The back compartment of the fire-box is arched over with fire-bricks placed at intervals apart: the combustion-chamber, also, is stocked with a faggot of perforated bricks placed at some distance clear of the tubes. Through these bricks the gases for combustion pass on their way to the tubes: they receive and retain a portion of the heat from the passing gases, when the fuel is incandescent and smokeless, and give it out to the smoke discharged from the fresh coal; thus acting as equalizers of temperature, for the purpose of consuming the smoke. The feed-water heating apparatus is in two parts, placed within and above the smoke-box; when in operation, steam from the blast-pipe is exhausted into the condenser above, where it meets the jet of cold water thrown in by one of the pumps, and is condensed by it; the feed-water, thus heated, flows back into the tender, except what is intercepted by the other pump, with which a junction is made from the return flow-pipe, for supplying the boiler. The water sent into the boiler passes, on its way, through the surcharging-chamber in the smoke-box, where it acquires a further increase of temperature, previous to entering the boiler. The surcharging-chamber contains a number of tubes for heating surface; it is supplied with heat from the exhaust steam, which is thrown directly into it from the exhaust-pipes, on its way to the condenser. Thus, when the engine is at work, the feed-water is heated, by the first process, to or near the boiling point, and by the second process to some point above that, having some relation to the initial temperature of the exhaust steam. The superfluous feed-water discharged from the condenser returns to the tender, and raises the general temperature within it.

The following are the particulars of the "Canute" passenger engine:—

Cylinder, 15 inches diameter, 21 inches stroke.
Driving-wheel, 6 feet 6 inches diameter.

Fire-box, inside dimensions:—
  Length, ... 4 feet 11 inches.
  Breadth, ... 3 „ 6 „
  Depth at back, ... 5 „ 1 „
  Do. at front, ... 4 „ 1 „
  Combustion-chamber, 3 feet 6 inches diameter, flat-roofed, 4 feet 2 inches long.
Tubes, 373 in number:—
  Outside diameter, ... 1¼ inch.
  Clearance between, ... ⁷⁄₁₆ „
  Length between Plates, ... 6 feet.
  Ashpans, entrance, 3 feet 6½ inches long, 12 inches deep at the front, and 9½ inches deep at the back.
Fire-grate, in two parts, 16 feet total area.
Heating Surface, interior:—
  Fire-box, ... 107 square feet.
  Combustion-chamber, ... 37 „
  Tubes, ... 625 „
  Total, ... 769 square feet.
Fire-bricks:—
  Gross Weight, 5¾ cwts.
  Heating Surface of bricks in contact with smoke, 80 square feet.

To find the ordinary quantity of heat contained in the fire-bricks when at work, one of each form of brick was heated to the usual degree of redness in an ordinary furnace, and plunged into a large vessel of cold water. The observed amount by which the temperature of the water was raised was multiplied into the quantity of water, and by the number of bricks of each form; and showed that the bricks contained upwards of 300,000 units of heat, equal to the heat derivable from 22 lbs. of carbon, which would evaporate 3 cubic feet of water.

The trials with the "Canute" were designed to test the performance of the engine with the usual express trains, and to measure the economical advantages of the fire-bricks as smoke-preventors, and of the feed-water heating apparatus. The engine was therefore tried under various conditions:—1st. In its usual order, with coal, bricks, and hot feed-water; 2nd. With coal, bricks, and cold water; 3rd. With coke, bricks, and hot feed-water; 4th. With coal, and hot feed-water, but without the bricks.

The coal used was of three kinds—Welsh, Griff, and Stavely. The first is rich in carbon, and is known as smokeless steam-coal. The second and third burn with considerable flame and smoke. The two kinds of coals were delivered to the engine usually in equal quantities, of one-half of Welsh coal and the remainder of Griff and Stavely coal. The water supplied to the engine was clean and hard; it was neutralized by an allowance of muriate of ammonia put into the tender.

The performance of the engine, with respect to the prevention of smoke, was completely successful. Under all circumstances, there was not at any time the slightest vestige of ordinary coal smoke from the chimney top. The engine did not require more than ordinary care and attention in manipulation, as the bricks were fully competent to the prevention of smoke.

To test the utility of the bricks as smoke-preventors, they were entirely withdrawn from the boiler in some of the trials; and without them, the smoke emitted was of a brown colour, and occasionally in such quantities as would certainly be called a nuisance. On restoring the bricks for the succeeding trials, the smoke vanished, and the light gases discharged from the chimney, when the steam was not admitted, had precisely the appearance of the vapour from burning coke. Economically, the bricks were found to be of utility, as the evaporative power of the coal was, by its more perfect combustion, materially greater with than without the bricks.

The trials with coke alone, as fuel, showed, by a marked contrast, the superiority of the arrangements of the boiler for burning coal, as the evaporative power and performance of coke of excellent quality was, contrary to ordinary experience, decidedly inferior to that of coal.

The trials with the feed-water heating apparatus shut off, pumping cold water from the tender in the usual way, caused a decided increase in the consumption of fuel.

There is much more that is worthy of remark in the results of these trials, but the limits assigned to this chapter prevent more than a brief summary of the results from being recorded. In the following summary, an allowance of 1¾ cwt. of fuel, for each double trip, is made for getting up the steam—equal to nearly 1 lb. per mile—and also a deduction for fuel and water consumed in running empty miles. *First.* With a regular express train and the engine in regular working order. The average train consisted of ten and a half carriages, estimated to weigh 66 tons, making, with the engine and tender, weighing 33 tons, a gross load of 99 tons. The average speed, exclusive of stoppages, was about thirty-four miles per hour. Whilst running, the quantity of water evaporated was at the rate of 82 cubic feet per hour of the time the steam was on the piston and the blast was in operation. The quantity of coal consumed was 547 lbs. per hour whilst under the blast. The consumption of coal was at the rate of 15 lbs. per train mile; and at the rate of ·15 lb. per mile per ton of gross load. The water evaporated per lb. of coal was 9·35 lbs. The average temperature of the feed-water was 187°.

Second. With a special weighted train, and the engine in regular working order, the train consisted of twenty-eight carriages, loaded to weigh 203 tons,—making the engine and tender 236 tons gross. The average speed, exclusive of stoppages, was 30¾ miles per hour. The quantity of water evaporated was at the rate of 130 cubic feet per hour, whilst under the blast. The quantity of coal consumed was at the rate of 915 lbs. per hour whilst under the blast; and at the rate of 28¾ lbs. per train mile, or ·122 per mile per ton of gross load. The water evaporated per lb. of coal was 8·87 lbs. The temperature of the feed-water was 212°.

Third. Recurring to the performances with the regular express trains, when the fire-bricks were withdrawn from the boiler, the consumption of fuel per ton per mile was ·17 lb.; and 1 lb. of coal evaporated 8·25 lbs. of water. The corresponding quantities with the brick were ·15 lb. and 9·35 lbs.,—showing a direct advantage of 12 per cent. in the consumption of fuel, and indirectly a similar advantage in its evaporative efficiency, due to the use of the bricks.

When coke was used instead of coal, the consumption was ·20 lb. per ton per mile; and 1 lb. evaporated 7·35 lbs. of water,—indicating in this particular class of engine a direct advantage in favour of coal of 24 per cent. in consumption, and indirectly an advantage of 21½ per cent. by evaporative efficiency.

When the feed-water heating apparatus was disconnected, the consumption was ·224 lb. per mile per ton; and 1 lb. of coal evaporated 8·24 lbs. of water,—indicating directly an economy of fuel due to the heating apparatus of 33 per cent. in consumption, and, indirectly, 12 per cent. of economy by increased evaporative efficiency.

The observed difference in evaporative efficiency introduces the subject of the damper, which is an important adjunct, materially affecting the efficiency of the fuel. The inferior evaporative efficiency observed when using cold feed-water is probably due to the dampers remaining wider open, in order to keep up the steam, and so passing an excess of air. It may be also due to some extent to the loss of surplus heat imparted to the heated feed-water in passing through the heated chamber inside the smoke-box.

Deducting the gain ascribed to the better use of the dampers from the total gain, the balance (21 per cent.) would represent the gain by the use of the heating apparatus, and by a little calculation would be shown to indicate a high ratio of efficiency. Thus, if the ordinary temperature at the time of the trials be taken at 50°, and the temperature of the exhaust steam be taken as that due to its mean pressure, which may be 20 lbs. per inch—as the usual initial pressure of exhaust was found by indicator-diagrams to be from 30 lbs. to 40 lbs.—the mean temperature of the exhaust steam in the body of the blast-pipe would be 260°; and the whole increase of temperature of the feed-water, on delivery to the boiler, would be (260° — 50°), or 210°. Now, the total heat of 100 lbs. of steam is 1,217°; and (1,217° — 50°), or 1,167°, is the whole amount of heat consumed in raising cold water of 50° into steam of 100 lbs. pressure. Then 1,167° ÷ 210° = 5·6 nearly, and the economy of fuel would be 1 — 5·6 lbs. of the total, or 18 per cent.

In comparing the performance of Mr. Beattie's engine with that of contemporaneous engines (in 1856), in which coal was used as fuel, the evaporative efficiency of the fuel in the latter might fairly be expressed by 6 lbs. of water per lb. of coal, whilst the former engine evaporated 9·35 lbs. of water per lb. of coal,—showing a superior evaporative efficiency of 3·35 lbs., representing an economy of fuel equal to 36 per cent. of the consumption by other systems.

Comparing Mr. Beattie's engine, then, as a coal-burning engine, with professed coke-burning engines, it was on a footing of equality with them in evaporative efficiency of fuel weight for weight.

By the employment of the feed-water heating apparatus, and by the more efficient use of the damper, in conjunction with that apparatus, an economy of fuel equal to 20 per cent. may safely be assumed, relatively to the consumption of fuel with cold feed-water.

The economical evaporative power of the boiler of the "Canute," in terms of grate area and the heating surface, is equal to 51 cubic feet of water per hour, according to the formula,

$$C = \cdot 00222 \left(\frac{h}{g}\right)^2,$$

deduced by the author from his observations with coke as fuel; in which C is the consumption of water in cubic feet per hour, at the rate of 9 lbs. per lb. of fuel; h is the total heating surface in square feet; and g is the grate area in square feet. A greater evaporative performance can only be attained through a partial sacrifice of fuel. It was accordingly found, on trial, that the "Canute," in evaporating, with excellent coke, 87 cubic feet of water per hour, which is considerably in excess of the economical maximum, 51 cubic feet, did so with a loss, at the inferior rate of 7·35 lbs. of water per lb. of coke. Whereas, by a remarkable contrast, water was evaporated in much greater quantities by coal, and with greater economy of fuel. Thus, 82 cubic feet of water per hour were evaporated at the rate of 9·35 lbs. per lb. of good coal, and 130 cubic feet per hour at the rate of 8·87 lbs. per lb. of coal.

IV.—Express Passenger-locomotive, by Messrs. Sharp, Stewart, & Co., Manchester. Plate XXIV.

Area of Grate, ...	14 square feet.
Heating Surface, ...	1,090 „
Diameter of Cylinders, ...	17 inches.
Stroke of Pistons, ...	24 „
Diameter of Driving-wheels, . .	6 feet 6 inches.
Length of Wheel-base, ...	15 feet.
Weight of Engine, in working order, .	29 tons.
Do. on Leading-wheels, . . .	9 „ 5 cwts.
Do. on Driving-wheels, . . .	10 „ 5 „
Do. on Trailing-wheels, . . .	9 „ 10 „
Do. of Engine, empty, . . .	26 „ 0 „
Do. of Tender, empty, . . .	11 „ 0 „
Do. do. full, . . .	20 „ 0 „

This engine is specially designed for heavy express traffic. It is constructed with a pair of single longitudinal frame-plates, carrying inside axle-bearings. The reversing gear is compactly balanced by special springs on the reversing shaft,—a system which possesses this advantage, that if the springs fail, there is no danger of derangement of the gearing; to the usual reversing gear is added a screw-motion, for the purpose of varying the degree of expansion to any desired degree of nicety,—the threads of the screw being in this instance substituted for the ordinary notches of the sector. The welcome cab is placed over the footpath for protection to the driver. The engine is fitted with Giffard's injector, conveniently placed at the end of the foot-plate. Wrought-iron axle-boxes are used for the driving and coupled axles, with the underhung springs. These are connected by a compensating beam, so disposed as to admit of the fire-box being made of the full width between the frame-plates.

CHAPTER XXIV.

EXAMPLES OF COUPLED PASSENGER-LOCOMOTIVE ENGINES.

THE coupled passenger engines illustrated, in addition to the coupled express engines of Mr. Beattie and Messrs. Sharp, Stewart, & Co., noticed in the previous chapter, are as follows:—

1. Coal-burning Passenger-locomotive, by Mr. J. J. Cudworth, Locomotive Superintendent, South-Eastern Railway. Plates XI. and XXVIII.
2. Passenger-locomotive with bogie, by the Rogers' Locomotive and Machine Works, Paterson, New Jersey, U.S. Plates XVIII., XIX., XX.
3. Passenger-locomotive with bogie, by Messrs. Robert Stephenson & Co., Newcastle-on-Tyne, for the Stockton and Darlington Railway. Plates XXVI., XXVII., XXVII*.
4. Passenger-locomotive, by M. Forquenot, for the Paris and Orleans Railway. Plate XXXIII.

I.—Coal-burning Passenger-locomotive, by Mr. J. J. Cudworth, for the South-Eastern Railway. Plates XI., XXVIII.

Area of Grate,	21	square feet.
Heating Surface,	965	„
Diameter of Cylinders,	16	inches.
Stroke of Pistons,	24	„
Diameter of Driving-wheels,	6	feet.
Length of Wheel-base,	15	„
Total Weight in working order,	30 tons	10 cwts.
Do. on Leading-wheels,	9 „	9 „
Do. on Driving-wheels,	10 „	15 „
Do. on Trailing-wheels,	10 „	6 „
Do. of Tender in working order,	20 „	10 „
Do. of Engine and Tender,	51 „	0 „

This engine represents the standard type of engine introduced in 1857, and matured by Mr. Cudworth, and employed on the South-Eastern Railway for passenger traffic. To Mr. Cudworth, locomotive engineers are greatly indebted for breaking up the routine practice of former years, in introducing and working out in a practical manner the long firebox with inclined grate, for the purpose of burning coal-slack.

This engine combines several good qualities. It has an excellent distribution of the load on the wheels, whereby 21 tons, or more than two-thirds of the gross weight of the engine, is placed upon the coupled-wheels. This desirable result is secured in virtue of the form of the boiler, which is made with a long firebox and a sloping grate, beneath which the hind coupled-axle is conveniently housed. Thus, with a length of wheel-base which is not excessive, is combined an extraordinarily long firebox, with a long run for the combustible gases, of which the hinder portion overhangs the hind axle. The firebox is 7 feet 6 inches long, and the grate is 7 feet long, having a short dead-plate at the upper end, and a clinker-plate for dropping the fire at the lower end, next the tube-plate. The firebox is divided longitudinally into two parts, by a water-space or diaphragm,—thus forming two distinct grates, each 18 inches wide. In the ordinary management of the fire, these grates are stoked alternately, small coal being burnt upon them, and the steam is easily and well kept up. The incandescent fuel slides forward, and the fresh coal is charged near the doorway. The aid of an auxiliary jet in the chimney is necessary for the combustion of smoke.

The following is a comparative statement of the results of the performance of the coal-burning engines of Mr. M'Connell, Mr. Beattie, and Mr. Cudworth, in 1854-60, borrowed from a paper written by the writer, and read at the Institution of Civil Engineers in 1860.[*] In Mr. M'Connell's engine, No. 303, of the London and North-Western Railway, there is a large grate, a long run, and air-tubes through the front and sides; and the system, though heavy on fuel, provides the greatest practicable area of firegrate, in order that the coal may burn with moderate intensity and free from clinker, and that plenty of air may pass into the firebox, through the grate, to consume the gases. The combustion is assisted by the introduction of air through the front and sides of the firebox; and a large capacity and length of run is provided, by adding a combustion-chamber, which is simply an extension of the firebox into the barrel of the boiler. To provide for alternate firing, the firebox is divided longitudinally into two compartments. This plan is, with good management, tolerably effective in preventing smoke, and was introduced in 1853.

Mr. Beattie's engine, the "Canute," has already been described (page 261); and amongst the engines specified in the table is to be found No. 118, which forms the subject of the illustrative plate of Mr. Cudworth's engines.

Of the three systems of extended firebars, by Mr. M'Connell, Mr. Beattie, and Mr. Cudworth, it appears, that with nearly equal gross weight—102 tons to 116 tons of engine, tender, and train—and at nearly equal speeds, Mr. M'Connell's system consumes 35½ lbs. of coal per mile, Mr. Beattie's 24 lbs. (the feed-water heater being shut off), Mr. Cudworth's 26 lbs.; or, per ton gross, Mr. M'Connell's evaporates 5·9 lbs. of water per lb. of coal, Mr. Beattie's 8·31 lbs., and Mr. Cudworth's 8·6 lbs. The excellence of Mr. Beattie's and Mr. Cudworth's systems is to be ascribed to the proximity of the radiant heating surface to the fuel and the flame, and in both systems the steam is well kept up.

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[*] "On Coal-burning and Feed-water Heating in Locomotive Engines" Minutes of Proceedings of the Institution of Civil Engineers, vol. xix., page 546.

COMPARATIVE PERFORMANCES OF COAL-BURNING ENGINES.

Designer.	Railway.	Engine.	Train.			Speed, &c.			Fuel.				Weather and Route.	Observer or Authority.	Date.
			Class.	Average Number of Vehicles.	Gross Weight of Engine, Tender, and Train.	Average Distance run between Stations.	Average Speed, including Stoppages.	Average Speed, excluding Stoppages.	Species of Fuel.	Gross Consumption.		Water Consumed per Lb. of Coal.			
										Per Mile.	Per Ton per Mile.				
					Tons.	Miles.	Miles per Hour.	Miles per Hour.		Lbs.	Lbs.	Lbs.			
Mr. M'Connell	L. & N.-W. R.	No. 303	Passenger	13·2	115	7·1	27·1	33·8	Hawksbury Coal	35·6	·31	5·9	Calm	Woods and Marshall	June and July, 1854
„	„	„	„	14	120	9·6	28·6	32·2	Coke	25·3	·21	8·9	„	„	„
Mr. Beattie	L. & S.-W. R.	Canute	„	11	102	15·7	29·6	33·0	Griff and Stavely Coal	24	·235	8·31	Strong side wind	Clark	March and April, 1856
„	„	„	„	11	102	14·3	30·1	33	„	17·4	·17	9·25	Strong side wind and calm. Heated feedwater, 131°	„	„
„	„	„	Special	28	236	39·4	28·8	30·7	„	30·5	·12	8·87	Slight wind. Heated feedwater, 212°	„	„
Mr. Cudworth	S.-E. R.	No. 142	Passenger	15	115	8	34·7	36·2	Coke	24	·21	8·2	Strong wind and calm. London and Dover Main Line	Cudworth	July, 1854
„	„	„	„	13·2	116	11	26·8	30·8	Hartley and Coking Coal	26	·225	8·6	Calm. London and Dover Main Line	„	Oct. and Nov., 1857
„	„	„	„	...	...	...	...	...	Coking Coal	21·26	...	8·3	Main Line	„	Two Years, ending March, 1860
„	„	Five Engines	„	...	...	...	...	...	Coke	26·36	...	...	„	„	„
„	„	Best of the five, No. 36	„	...	...	...	...	...	„	21·86	...	...	„	„	„
„	„	Nos. 173, 174	Goods	...	...	...	...	...	„	31·83	...	...	„	„	One Year, ending March, 1860.
„	„	Nos. 108, 111	„	...	...	...	...	...	Coking Coal	29·77	...	8·2	„	„	„
„	North Kent	Five Engines	Passenger	...	...	...	...	...	Coke	25·3	...	...	London and Maidstone	„	Jan. to March, 1860
„	„	No. 5	„	...	...	...	...	...	Coking Coal	25·44	...	8·9	„	„	„
„	„	No. 159	„	...	...	...	...	...	Coke	25·22	...	...	„	„	„
„	„	No. 131	„	...	...	...	...	...	Coking Coal	24·58	...	8·9	„	„	„
„	„	No. 181	„	...	...	...	...	...	Coke	32·08	...	...	„	„	„
„	„	No. 118	„	...	...	...	...	...	Coking Coal	23·01	...	8·9	„	„	„

II.—Passenger-locomotive, by the Rogers' Locomotive and Machine Works, Paterson, New Jersey, U.S. Plates XVIII., XIX., XX.

Area of Grate,	13 square feet.
Heating Surface,	787 „
Diameter of Cylinders,	16 inches.
Length of Stroke,	24 „
Diameter of Driving-wheels,	5 feet 8 inches.
Length of Wheel-base, extreme,	20 „ 1½ „
Do. to centre of Bogie,	23 „ 0 „
Do. of Bogie alone,	5 „ 9 „

This engine represents, as a type, the one general form of passenger engine now retained in North America, although a great variety, and often extraordinary types, of engines have existed there. In his paper on "American Locomotive and Rolling Stock," Mr. Zerah Colburn supplies the following general description of the modern American passenger engine:—"It has in most cases outside-cylinders—indeed, inside-cylinder engines have not been built for many years—and it invariably has four coupled driving-wheels, and a four-wheeled bogie. As a general representation of all the passenger engines now built, it may suffice to describe one constructed at the workshops of the Chicago and North-Western Railroad; and although having driving-wheels but 5 feet 8 inches in diameter, it has run a special train, probably of no great weight, ninety-one miles in ninety-five minutes, of which run fifty-one miles were made in forty-nine minutes, as certified by Mr. Cushing, Locomotive Superintendent of the line. High speeds are not, however, maintained on American lines, and no express train is timed to more than thirty-two miles in an hour, nor to more than twenty-seven miles an hour in a run of 300 miles, although in this case the stops are longer and more frequent than on English lines. The speeds of passenger, and even so-called express trains, are more generally from twenty to twenty-five miles an hour, while average speeds of from fourteen to eighteen miles an hour are scheduled in the time-tables of many local lines in the Western and Southern States. Nor does the American Bradshaw (*Appleton's Guide*) show a single instance of a continuous run without stopping of more than forty miles; and those of even thirty-two to thirty-six miles are unfrequent, most of the so-called expresses stopping at more frequent intervals, often at distances of from ten to fifteen miles. For such service an adhesive weight of from 14 to 16 tons, moderate-sized driving-wheels and cylinders, and the means of working full steam-pressure for nearly the whole stroke in starting, are all that is required. The leading dimensions of the representative type of passenger engines are,—cylinders from 15 inches to 17 inches in diameter, with a length of stroke of from 22 inches to 24 inches, and coupled driving-wheels of from 5 feet to 5 feet 8 inches in diameter. Such engines will exert a tractive force of 3½ tons to 4 tons in starting, for which their adhesive weight, assisted sometimes by sand, is sufficient; and thus they can get quickly away from stations even with trains of a gross weight of 200 tons or more. Economy of fuel has not been studied to the same extent in American as in English locomotives: the blast-pipes of the former are smaller, the draught more forced, the back pressure greater, and less expansion is attempted in the cylinders, the link-motion being generally arranged to cut off at one-third stroke as a minimum, and nine-tenths or more as a maximum. It is thus that boilers of moderate size are made to supply steam for work equal to 300 indicated horse-power, or the exertion of upwards of 2 tons of draught upon a passenger train at a mean speed of twenty-five miles an hour; but there is nothing remarkable in the consumption of from 50 lbs. to 60 lbs. of coal per mile in such work."

"As relating to the consumption of fuel, the Pennsylvania Central Railroad may be instanced. Its main line is 356 miles long, the eastern and western portions having gradients of 1 in 100, while its middle division has but very moderate ascents; for a few miles, however, on the western division, there are gradients of 1 in 55, up which the goods trains are assisted by pushing engines. Of 409 engines on the line, in October, 1868, 240 had six coupled, and three had eight coupled-wheels, of from 3 feet 8 inches to 4 feet 6 inches in diameter, the larger number having 18-inch cylinders with 22-inch stroke. The standard pattern of passenger engines had 17-inch cylinders, 24-inch stroke, and four coupled-wheels, 5 feet to 5 feet 6 inches in diameter,—and of this class there were upwards of 100. The mean speed of fast passenger trains is about twenty-five miles an hour, and of other passenger trains about twenty-three miles an hour; yet the consumption of coal for both goods and passenger engines averaged respectively,—On the eastern, middle, and western divisions, 52·7 lbs., 66¾ lbs., and 70⅝ lbs. per engine-mile, corresponding to a general average of about 70 lbs. per train-mile, the goods mileage being about three and a half times the passenger mileage. The coal is of excellent steaming quality, and is found so near at hand, and in such abundance, that its cost is after all but about 3⅜*d.* in currency, or 2¼*d.* in coin, per train-mile. At this rate a difference in consumption of 24 lbs. of coal per mile would only cause a variation of a penny per mile in the cost of fuel; and it has been argued that such a waste is better than the alternative of employing an engine 4 tons or 5 tons heavier, to work with a less rapid rate of combustion, a slower piston-speed, and more expansively. It is to be remembered that for every lb. of coal burnt, from 6 lbs. to 7 lbs., or 8 lbs. of water require to be provided, and an additional weight of at least half as much in the structure of the tender itself is necessary to carry it. Thus, if an engine burn 67 lbs. of coal, and consume seven times the weight, or 469 lbs. of water per mile, and stops every thirty-three and a half miles, it will require to start with at least a ton of coal, and 7 tons of water,—the average weight of both progressively consumed over the whole distance being 4 tons. If the coal could be economized one-fourth, the average weight of coal and water would still be 3 tons; and it appears better, therefore, where fuel is so cheap, to waste a portion of it—perhaps a considerable portion—than to economize it by adding 4 tons or 5 tons of weight to the engine, to obtain a larger boiler, and working gear admitting of a slower piston speed and a higher grade of expansion. The question turns upon the cost per ton per mile of moving a passenger engine and tender, but much more upon the length of run without stopping; upon whether the water may be picked up along the line, or taken only at stations; and upon the original cost and the interest on it, the repairs, and the depreciation of the engine. Every locomotive engineer can easily examine the question in the light of these circumstances as applying to his own line; and it might be found in some cases that it would be absolutely cheaper to employ engines burning 40 lbs. of coal per mile, than heavier and costlier engines burning about 30 lbs."

"Much of what has just been observed of the passenger engines," proceeds Mr. Colburn, "applies with even greater force to the goods engines. Of but moderate total weight, they have large cylinders and small wheels, and they draw heavy trains at a fair speed with a consumption of coal often amounting to 100 lbs. or more per mile. One instance, taken from the regular working of the standard type of goods engine on the Pennsylvania Central Railroad, will suffice. This class of engines has ten wheels, of which six, each 4 feet 6 inches in diameter, are coupled. The whole weight of the engine is only 31½ tons, and of this but 23½ tons rest on the coupled-wheels, available for adhesion. The cylinders are 18 inches in diameter, with a length of stroke of 22 inches. With 60 lbs. mean effective pressure per square inch upon the pistons, these engines would exert a tractive force, less their own internal resistances, of rather more than 3½ tons, or about one-seventh of their adhesion-weight, although in starting a train or in ascending a gradient with 100 lbs. pressure on the pistons, the steam tractive force would be 6 tons, equal to more than one-fourth of the adhesion weight, the efficiency of which would then be assisted when necessary by sand. There are thirty of this class of engines, besides many others of nearly the same weight and dimensions, on the middle division alone of the Pennsylvania Central line, this division being 132 miles long, and gently descending for nearly its whole length to the eastward, the only exceptions worth noticing being 300 yards up 1 in 293, and 400 yards up 1 in 352. The regulation load going eastward, in which direction only the waggons are nearly or fully loaded, is forty-five eight-wheel waggons, each weighing 7½ tons empty, and 16½ tons loaded, making a total load 743 tons. The total fall in the 132 miles is 851 feet, equivalent to an average descent of 1 in 808 for the whole distance. Yet, in October last, 1868, the average consumption of the best bituminous coal by these engines was 100 lbs. per mile, in addition to the wood employed for lighting, and the average for all engines, passenger and goods, 114 in number, was 66⅔ lbs. per mile. On the same division, two engines having six wheels coupled, and a two-wheel bogie, the whole weight being 31¾ tons, of which all but 2¼ tons were available for adhesion, were loaded to heavier work. They had 19-inch cylinders, 24-inch stroke, and 4-feet coupled-wheels, and with 60 lbs. mean effective cylinder pressure would exert a tractive force, less their own internal resistances, of 4·83 tons, equal to about one-sixth of the adhesion weight. The distance run by these engines averaged 100 miles a day each, with a consumption of coal respectively of 127½ lbs. and 104 lbs. per mile. Their regulation load going eastward was sixty eight-wheeled waggons, weighing, when fully loaded, 990 tons. It is not probable, however, that they were always fully loaded, nor, even if they were, can an accurate estimate of the resistances be formed. Yet, if the latter be taken, after allowing for gravity in favour of the eastward trip, at 8⅝ lbs. only per ton for a train, engine and tender included, of 1,040 tons weight, the total resistance would be 4 tons, and thus the work done on each mile would be equal to twenty-four horse-power exerted for one hour. For—

$$\frac{5{,}280 \text{ feet per mile} \times 8{,}960 \text{ lbs. (4 tons)}}{1{,}980{,}000 \text{ foot-lbs. per horse-power}} = \left\{ \begin{array}{l} 24 \text{ horse-power} \\ \quad \text{per hour} \end{array} \right.$$

nearly; so that, could these estimates be trusted, the consumption of coal per horse-power per hour would not after all exceed 4¼ to 5¼ lbs. The speed would not probably exceed fifteen miles per hour, corresponding to 360 horse-power. Exact data are unfortunately wanting to determine the dynamical effect of these engines; and it is not pretended that the general assumptions just made have any claim to exactness, although the author is convinced, from a somewhat lengthy acquaintance with engines of the same general type, that the estimates above given are not very far wide of the mark."

In addition to the bogie and the cast-iron chilled wheels, the framing constructed of forged bars is conspicuous, forming a strong contrast to the slab frame of English engines. It does not materially differ from the framing originally introduced and adopted by Messrs. Bury, Curtis, & Kennedy in England; the bogie is constructed on the Bissell system, which has already been described (page 98). The valve-gear constructed with a rocking shaft, shown in detail in Plate XX., through which the action of the link-motion is transmitted laterally to the slide-valve, which is placed vertically over the cylinder on a horizontal face. Thus it is that the Americans provide the means of working the slide-valves of outside cylinders with outside valves:—They work from eccentrics and link-motions placed inside the axle bearings and the frame, through a rocking shaft, to the valve outside. Continental engineers, on the contrary, for the most part, derive the valve-motion direct from eccentrics fixed on an overhung crank outside the frame and crank-pin. In the Rogers' engine, the reversing gear is balanced by a laminated spring suspended from the framing.

III.—Passenger-locomotive, by Messrs. R. Stephenson & Co., Newcastle-on-Tyne, for the Stockton and Darlington Railway. Plates XXVI., XXVII., XXVII*.

Area of Grate, ...	12¾	square feet.		
Heating Surface, ...	1,053	„		
Diameter of Cylinders, . . .	16	inches.		
Length of Stroke, ...	24	„		
Diameter of Driving-wheels, . .	7	feet	½	inch.
Length of Wheel-base, extreme, . .	20	„	¼	„
Do. to centre of Bogie,	16	„	11¾	„
Do. of Bogie alone, .	6	„	1	„

This engine with large wheels properly belongs to the class of express engines, but it has not been so denominated by the constructors. The axles of the bogie of this engine are placed a good distance apart,—at 6 feet 1 inch centres. In the absence of the controlling radial action of the Bissell system, the provision of a long wheel-base for the bogie is the best means of insuring safe action in entering and going along curves. A small auxiliary tank for heating the feed-water, by means of the spare steam from the boiler, is placed below the foot-plate.

IV.—Passenger-locomotive, by Mr. Forquenot, for the Paris and Orleans Railway. Plate XXXIII.

Area of Grate, ...	14·5 square feet.
Diameter of Cylinders, ...	16½ inches.
Length of Stroke, ...	25⅝ „
Diameter of Driving-wheels, . . .	6 feet 7 inches.
Length of Wheel-base, ...	13 „ 1½ „
Total Weight of Engine, in working order,	33 tons 12½ cwts.
Do. on Leading-wheels, . .	8 „ 17½ „
Do. on Driving-wheels, . .	12 „ 5 „
Do. on Trailing-wheels, . .	12 „ 10 „

This engine represents a numerous class of engines employed on the Paris and Orleans Railway, for passenger traffic. The firegrate is inclined, and in two sections, of which the lower is moveable,—the total length of grate being 4 feet 4½ inches. A water-space partition is carried across the firebox, sloping like the grate, with the object of mixing the coal-smoke with air, entering at the doorway. The cylinders are outside, and the valve-gearing is outside, and worked by overhung eccentrics. The steam-dome is very large, being 3 feet in diameter and 3 feet 6 inches high. There are four coupled-wheels, and the firebox is overhung. A screw-brake is applied to the engine, worked from the foot-plate. This engine is not to be recommended as an example to be imitated. In design and detail it is defective. There are many useless refinements, and there is generally an absence of simplicity and direct action.

CHAPTER XXV.

EXAMPLES OF GOODS-LOCOMOTIVE ENGINES.

1. Goods-locomotive, by Messrs. Sharp, Stewart, & Co., Manchester, for the Egyptian Railways. Plate V.

2. Goods-locomotive, by Messrs. Neilson & Co., Glasgow, for the Valencia and Ponferrada Railway, Spain. Plates XII., XIII., XIV.

3. Goods-locomotive, by Mr. John Ramsbottom, for the London and North-Western Railway. Plate XV.

4. Mixed-traffic locomotive, by Messrs. Beyer, Peacock, & Co., Manchester, for the Smyrna and Cassaba Railway. Plates XXIX., XXX., XXXI., XXXII.

5. Goods-locomotive, by Mr. John Wakefield, Dublin, for the Great Southern and Western Railway of Ireland. Plates XXXIV., XXXV.

6. Goods-locomotive, by Messrs. J. F. Cail & Co., Fives-Lille, for the Great Northern Railway of France. Plate XL.

7. Goods-locomotive, by Mr. William Cowan, Locomotive Superintendent, Aberdeen, for the Great North of Scotland Railway. Plates XLI., XLII.

8. Mixed-traffic locomotive, by Messrs. Fox, Walker, & Co., Bristol, for the Windsor and Annapolis Railway, Nova Scotia. Plate XLVI.

The lines of demarcation between goods, mixed, and passenger engines are not now so definite as formerly. In the classification of engines of which examples form the subjects of illustration, the denominations applied to them by the designers or constructors have been adopted in these descriptive chapters. Several of the engines described in this chapter as goods engines might have been classed as express or passenger engines.

I.—Goods-locomotive, by Messrs. Sharp, Stewart, & Co., Manchester, for the Egyptian Railways. Plate V.

Area of Grate, ...	27·5 square feet.
Heating Surface, ...	1,192 „
Diameter of Cylinders, . . .	17 inches.
Stroke of Pistons, ...	24 „
Diameter of Driving-wheels, . .	5 feet.
Length of Wheel-base, . . .	15 feet 6 inches.
Weight of Engine, in working order, .	32 tons.
Do. on Leading-wheels,	10 „ 12 cwts.
Do. on Middle-wheels, .	11 „ 11 „
Do. on Trailing-wheels,	9 „ 17 „
Weight of Engine, empty, . . .	28 tons 15 cwts.
Do. of Tender, empty, . . .	11 „ 0 „
Do. do. full, . . .	20 „ 0 „

This engine, the "Manchester," though sold to His late Highness Said Pacha, Viceroy of Egypt, for the Egyptian Railways, was one of a number of similar engines constructed by the same manufacturers for the London, Chatham, and Dover Railway. It bears a high reputation on the Egyptian Railways; and with respect to the Chatham line, this class of engine, with 5-feet wheels and 17-inch cylinders, of maximum power, are adapted for the heavy loads, steep gradients, and high speeds of that line.

The engine was exhibited in 1862, and was of excellent workmanship and magnificent balance and proportions,—for which last it is greatly indebted to the long firebox, with inclined grate, adopted from the practice of Mr. Cudworth. It weighs, empty, 28¾ tons, full, 32 tons; yet the distribution of this ponderous load is, by the happy distribution of material, nearly equalized over the three axles. The firebox is 8 feet long externally, with an inclined grate, on Mr. Cudworth's system, for burning coal, which has already been described at length. The hind axle is thus admitted under the firebox about one-third of its length, and is enabled to take its proper share of the load, whilst the length of wheel-base is moderated, and is bisected by the middle axle,—advantages peculiar to Mr. Cudworth's system. The firebox measures 7 feet 3¼ inches long inside, divided longitudinally by a "mid-feather," and has 27½ square feet of grate. There are 189 flue-tubes, 2 inches diameter, and ⅜ inch clear, within a 4 feet 2 inch barrel. These are good proportions, according to current practice; but had there been only 160 tubes at ¼-inch clearance in such a large barrel, it would have been better. The boiler is fitted with Giffard's injectors; and it has a lock-up safety-valve on the firebox shell, in addition to the safety-valves on the dome. The housing for the enginemen, with windows in the front and sides, is very good and complete.

The frame is composed of four longitudinal plates, carried from end to end, with suitable cross-plates to bind it. The outer plates are each of one thick slab, of considerable depth between the guards, with intervals slotted out, combining great strength and lightness. In this engine, the usual construction of outer frame with double plates and timber packing has thus been superseded. The driving-axle has four bearings, two inside and two outside, and the extreme axles have only outside bearings; thus the old antagonistic action is set up, arising out of the unequal wear of inside and outside bearings,—the inside bearings next the crank, receiving the full strain of the steam, wearing faster than the outsides, leaving the cranks unsupported, and shortening the duration of the axle; overstraining in that way, also, and loosening the connections of the inner and outer frame-plates. On the contrary, this arrangement admits of large outside bearings and capacious axle-boxes for all the axles,—an advantage which unfortunately the narrow gauge does not permit for inside crank-axle bearings. The drawing-tackle of this engine is connected exclusively to the frame.

The valve-gear is very substantially and firmly arranged.

The expansion-link is shifting, and the slide-block is not overhung, but hung between the ends of the link which carries it.

The "Manchester" is said to be capable, with a working pressure of 120 lbs. per inch, of taking a load of 480 tons on a level, at twenty miles per hour, or a load of 250 tons up a gradient of 1 in 100, at fifteen miles per hour.

It may be useful to compare this engine with the six-coupled engine which was exhibited alongside, of the same weight and calibre, constructed by Messrs. Wm. Fairbairn & Sons, Manchester, for the Midland Railway, to the designs of Mr. Kirtley, the Locomotive Superintendent. The Midland engine has the ordinary rectangular firebox, and is fitted on the Midland system, with a fire-brick arch and baffle-plate, for burning coal. It has 14½ square feet of grate; the cylinders are 16 inches diameter, with 24 inches stroke. The engine weighs, in working order, 32 tons 5 cwts. It is a thoroughly good engine of its kind, and is a type of the powerful goods engines usually met with on the old main lines of railway. But in the first place, the weight is not so well distributed on the wheels as in the "Manchester," being considerably in excess on the leaders, thus—

	Midland.		"Manchester"	
	Tons.	Cwts.	Tons.	Cwts.
Leading-wheels, ...	12	6	10	12
Middle-wheels, ...	11	5	11	11
Trailing-wheels, ...	8	14	9	17
Total Weight, ...	32	5	32	0

Showing, in the Midland engine, 3 tons 12 cwts. more on the leading than the trailing-wheels, and in the "Manchester," only 15 cwts. more weight; the wheel-base of the Midland engine being 16 feet 6 inches, or 1 foot more than that of the other, as the vertical firebox keeps back the hind axle. The frame of the Midland engine, like the other, has four longitudinal plates, with four bearings to the driving-axle, and outside bearings to the other axles; but the inside frame-plates stop short at the firebox, and are fastened to it by sliding joints, allowing for expansion of the boiler. The draw-plates are rivetted to and across the back of the firebox, from which the whole of the drag is taken; and, of course, as the tractive force must be transmitted from the outside frame-plates to the firebox, they are very strongly united by brackets to the firebox, for that purpose. Thus the whole area of the frame behind the firebox remains unemployed, except as standing ground. This is an objectionable feature in the Midland engine, without any countervailing advantage. The steam-power is not only transmitted circuitously, tending to overstrain and buckle the framing, but it arbitrarily subjects the boiler, already highly strained, to an unnecessary additional fatigue, which is now beneficially avoided in other and recent practice. The flue-tubes are well designed, being 2 inches in diameter, 180 in number, and placed at ⅝-inch clearance, in vertical rows, which accounts for the high character of the Midland engines for keeping up the steam. The boiler is put together without angle-iron, the plates being flanged at the end; the steam-dome also is flanged and formed in one piece. For this purpose, Alton & Fernie's patent thick-edged plates are used; they are rolled to ⅝-inch and ¾-inch thick, to allow material to work upon. The advantages claimed for the thick-edged plates in their application are, that they save a joint, including a row of rivets, and are stronger than angle-iron joints; the joints are cheaper and easier to make, and are not subject to grooving; the dome-joints being faced in the lathe, bad joints are prevented.

II.—Goods-locomotive, by Messrs. Neilson & Co., Glasgow, for the Valencia and Ponferrada Railway, Spain. Plates XII., XIII., XIV.

Area of Grate, ...	15·5 square feet.
Heating Surface, ...	1,551 „
Diameter of Cylinders, ...	17 inches.
Stroke of Pistons, ...	24 „
Diameter of Driving-wheels, . .	4 feet.
Length of Wheel-base, . . .	10 feet 6 inches.
Weight of Engine, in working order, .	34 tons 15 cwts.
Do. do. empty, . . .	31 „ 0 „
Do. of Tender, loaded, . . .	17 „ 10 „
Do. do. empty, . . .	9 „ 10 „

This is a well-designed engine, adapted for working on steep gradients. It was constructed to run on the 5 feet 6 inch gauge of rails; and full advantage has been taken of the greater width of gauge as compared with the ordinary or "narrow" gauge (4 feet 8½ inches), in providing large boiler-room and bearing surface on the axles. The firebox is 3 feet 11¼ inches wide inside, and 4 feet 1 inch long, giving from 15 to 16 square feet of grate-area. The barrel of the boiler is telescopic, being 4 feet 4 inches in diameter at the firebox, and 4 feet 1 inch next the smokebox, made of ½-inch plates. There are 207 flue-tubes, 2 inches in diameter outside, placed at 5/16 inch apart; and as the barrel of the boiler is 13 feet 3 inches in length, and affords a flue-tube 13 feet 6½ inches in length, a large heating surface, amounting in all to 1,551 square feet, is obtained. There is reason to believe, however, that the tube-surface is considerably overdone. The tubes are too closely spaced; and if they had been fewer in number and more widely apart, so as to give a total heating surface of about 1,000 square feet, the evaporative efficiency of the engine would have been better. As it is, there can be no question of the unlimited steaming power of the engine. The axles have plenty of bearing surface, which, in a dry, dusty country, is of the greatest utility. The journals are 9 inches long. The engine is of a most powerful class, and has a compact wheel-base,—only 10 feet 6 inches. The tender holds 1,320 gallons of water.

III.—Goods-locomotive, by Mr. John Ramsbottom, for the London and South-Western Railway. Plate XV.

Area of Grate, ...	15 square feet.
Heating Surface, ...	1,102 „
Diameter of Cylinders, ...	17 inches.
Stroke of Pistons, ...	24 „
Diameter of Driving-wheels, . . .	5 feet 2 inches.
Length of Wheel-base, ...	15 „ 6 „
Total Weight of Engine, in working order,	27 tons.
Do. on Leading-wheels, . .	9 „ 14 cwts.
Do. on Driving-wheels, . .	10 „ 0 „
Do. on Trailing-wheels, . .	7 „ 6 „
Do. of Tender, in working order,	21 „ 8 „

This engine is of the ordinary type of six-coupled goods engine, with single frame-plates and inside bearings. The firebox, like that of the "Lady of the Lake," is fitted for smoke prevention, with a brick-arch and two 7-inch square openings, through the front of the box, for the admission of air above the fuel. The reversing gear is fitted with a screw motion, and the boiler is fed by means of an injector, and is fitted with Mr. Ramsbottom's duplex safety-valve. There are 192 flue-tubes, 1¾ inches in diameter and 10 feet 9 inches long. The firegrate is 4 feet 2⅝ inches long, and 3 feet 6¾ inches wide. The engine takes an average train, weighing 217 tons, at a speed of 19·41 miles per hour, with a consumption of fuel equal to 42·19 lbs. per mile.

IV.—Mixed-traffic locomotive, by Messrs. Beyer, Peacock, & Co., for the Smyrna and Cassaba Railway. Plates XXIX., XXX., XXXI., XXXII.

Area of Grate, ...	14·9 square feet.
Heating Surface, ...	1,045 "
Diameter of Cylinders, ...	16 inches.
Stroke of Pistons, ...	22 "
Diameter of Driving-wheels, . .	5 feet.
Length of Wheel-base, ...	15 "

This engine has four coupled-wheels under the boiler, with single trailing-wheels. It is obviously an engine capable of great execution, with free running qualifications—the freedom of running being promoted by the separation of the hind-wheels from the coupled-wheels. It is fitted for burning coal with the ordinary brick-arch and baffle-plate in the firebox. The springs over the front and middle axles are connected by balance levers, which for a four-coupled engine is a most desirable adaptation. For additional remarks, see the next notice.

V.—Goods-locomotive, by Mr. John Wakefield, for the Great Southern and Western Railway of Ireland. Plates XXXIV., XXXV.

Area of Grate, ...	14·7 square feet.
Heating Surface, ...	1,190 "
Diameter of Cylinders, . .	16 inches.
Stroke of Pistons, . . .	24 "
Diameter of Driving-wheels, . .	5 feet 2 inches.
Length of Wheel-base, . . .	15 " 2 "

This engine, like the one just noticed, has four coupled-wheels under the boiler, and single trailing-wheels. It has only 178 flue-tubes, as against 191 tubes in Messrs. Beyer, Peacock, & Co.'s engine; but they are 2 inches in diameter, and 11 feet 10½ inches long, as against 1¾ inches diameter, and 10 feet 3 inches long. Notwithstanding the longer tube and barrel of boiler in Mr. Wakefield's engine, his wheel-base is only 2 inches longer than Mr. Beyer's; and he arrives at this partly by putting in his firebox 5 inches shorter than Mr. Beyer's, and extending it laterally, taking the benefit of the greater facilities of the 5 feet 3 inch gauge of the Irish Railways, and thus obtaining as large a grate as in Mr. Beyer's engine. In presence of the greater width of gauge, there is less reason for confining the water-spaces about the firebox to a width of 3 inches,—it is only half an inch more than that of Mr. Beyer's water-spaces. A 4-inch water-space all round would have much improved the circulation of the water and the evaporative efficiency of the firebox.

Again, there is a considerable degree of obliquity of action in the valve-gear in Mr. Wakefield's engine, which could easily have been avoided. Whilst the valve-spindles are only 3¼ inches apart to their centre lines, the eccentrics, at the origin of the motion, are 8 inches apart to their centre lines, showing a total obliquity measured by 4¾ inches, or 2¼ inches out of the centre line, for each motion. Whereas, in Mr. Beyer's engine, the eccentrics are only ⅜ths of an inch out of the centre line of the relative spindles. One immediate cause of the divergence in Mr. Wakefield's engine is the requirement for space between the valve-rod links to clear the counter-weight on the reversing shaft, which is placed centrally,—a necessity which Mr. Beyer avoids by employing two counter-weights, one for each link.

The greater freedom laterally for arranging the machinery in Mr. Wakefield's engine, is indicated by the width apart of the frame-plates—4 feet 7 inches—as compared with 4 feet 2 inches in Mr. Beyer's engine; and Mr. Wakefield takes the benefit of it in putting in crank-arms 5 inches in thickness, which Mr. Beyer only makes 4 inches thick. The suspending links, which carry the valve-rod links, are hung from the boiler by Mr. Wakefield, though there is a transverse frame-plate not far off, which might have been adjusted to carry the links. This Mr. Beyer actually does, leaving the boiler untouched. But Mr. Wakefield, it may be said, wanted to attach his feed-pumps to the transverse frame-plates, which could not, with this object in view, well be placed further back nearer to the suspending links. Mr. Beyer, however, shows how to carry the pumps, by fixing them entirely to the longitudinal plates, though, no doubt, Mr. Wakefield's pumps lie an inch further off from the longitudinals than Mr. Beyer's. The side frame-plates of Mr. Wakefield's engine stand very high: they are 4 feet 2 inches above the rails; whereas Mr. Beyer's are 5 inches lower. The superiority of the lower level is manifest,—it places the frame-plate more nearly central with the cylinders, axles, and other working parts attached to them. The pumps are better fixed; and the reversing shaft—which in Mr. Wakefield's engine abuts against the frame-plate, and consequently requires the aid of a distorted lever, reaching over the edge of the frame-plate, to take the motion of the reversing-rod—is, in Mr. Beyer's engine, carried over the edge of the frame-plate, and is controlled by a straight radial lever. The disadvantage of the high-set frame-plates is further manifested in the fixings of the cylinders, which Mr. Wakefield carries by means of wings thrown upwards to the side-plates, out of the centre line; whilst Mr. Beyer plants them solidly on the side-plates by two strong ribs above and below the centre line.

The two engines above in part compared, afford useful examples of what to imitate and what to avoid,—they exemplify direct and indirect action.

VI.—Goods-locomotive, by Messrs. J. F. Cail & Co., Fives-Lille, for the Great Northern Railway of France. Plate XL.

Area of Grate,	25 square feet.
Heating Surface,	2,200 ,,
Diameter of Cylinders,	19¼ inches.
Stroke of Pistons,	$25\frac{5}{16}$,,
Diameter of Driving-wheels,	4 feet 3½ inches.
Length of Wheel-base,	13 ,, 10¾ ,,
Weight of Engine in working order,	43 tons.
Do. empty,	37 ,,

This engine has outside cylinders, with the valves and valve-gear outside the cylinders and crank-pins. The fire box is constructed for burning coal, on the system of M. Tenbrinck, with a long sloping grate, and a water partition, or mid-feather, across the firebox, above and nearly parallel to the grate, so as to deflect the flame towards the door, and meet the air admitted there,—combining some of the features of Mr. Cudworth's and Mr. Beattie's systems of firebox. It is needless to repeat that the enormous expansion of heating surface to 2,200 square feet is only arrived at by packing an uselessly large number of tubes into the boiler, and defeating the object aimed at, cf great evaporative power. The reversing-rod is stayed at the middle of its length by a friction-roller working on a segmental guide.

VII.—Goods-Locomotive, by Mr. William Cowan, Aberdeen, constructed by Messrs. Neilson & Co. for the Great North of Scotland Railway. Plates XLI., XLII.

Area of Grate,	13 square feet.
Heating Surface,	
Diameter of Cylinders,	16 inches.
Stroke of Pistons,	24 ,,
Diameter of Driving-wheels,	5 feet 6½ inches.
Length of Wheel-base, extreme,	20 ,, 5 ,,
Do. to centre of Bogie,	17 ,, 5 ,,
Do. of Bogie,	6 ,, 0 ,,
Total Weight of Engine in working order,	35 tons 9 cwt.
Do. on Bogie-wheels,	12 ,, 6 ,,
Do. on four-coupled wheels,	23 ,, 3 ,,
Do. of Engine, empty,	31 ,, 12 ,,
Do. of Tender, empty,	10 ,, 1 ,,
Do. do. full,	16 ,, 10 ,,

This engine is the latest development of the primary type of locomotive originally designed, in 1854, by the writer, and constructed by Messrs. William Fairbairn & Sons for the Great North of Scotland Railway,—having four-coupled wheels, coupled behind, and a pair of leading-wheels.* These first engines were designed with a moderately-sized firebox, for burning coke as fuel, on the principle advocated by the writer, as the result of much observation and considerable experience, that a fire-grate, just large enough, and no larger than was sufficient for the proper combustion of coke, was more efficient than a larger grate. In those first engines the grate was limited to a length of 3 feet and an area of 10½ square feet. The flue-tubes, likewise, were comparatively few, having been limited to 154 in number, and to 1¾ inches in diameter, in order to afford plenty of space about and amongst the tubes for the generation and liberation of steam and the free circulation of water. Subsequent experience, and the best practice of the present day, have confirmed the correctness of the writer's views with regard to the necessity and advantage of free space for circulation about the tubes. Continental locomotists, however, have yet to perceive the vanity of accumulating tube-surface, regardless of the circulation. They pack their boilers with flue-tubes, though they have not succeeded in getting anything more out of their engines on that account. On the contrary, with respect to the area of the fire-grate, experience has shown that a larger area may with advantage be employed for the combustion of coal than for that of coke. At the same time, the principle of a limited area of grate is common to the combustion and evaporative performance of coke and of coal as distinguished from slack; and it may be observed, that in the latest design of Mr. Cowan, illustrated in the engravings now under notice, whilst he has considerably enlarged the area of the grate, as compared with that of the earliest engines, in response to the requirement of greater surface for burning coal—having extended it to 13 square feet—still, it is less, in relation to the dimensions of the cylinders and wheels, than that of any other engine exemplified in this work. In the primary type of engine, it may be added, a heavy mass of cast-iron was placed under the foot-plate, behind the firebox, for the purpose of increasing the adhesion-weight on the hind wheels, as the writer considered that it was better to apply auxiliary dead weight cheaply, than to apply it in the form of heavier frames and fireboxes, costing ten times as much. Again, in the primary type, the leading axles, as also the axles of the tender, were extended considerably beyond the wheel on each side, so that an element of elasticity might be brought into play, as between the wheel, which is subject to concussions, and the journal or axle-bearings, which considerably overhang the wheels. It is commonly supposed that the nearer an outside-bearing is brought to the nave of the wheel, the better; but, inasmuch as it thus becomes more rigid, and is necessarily, in respect of rigidity, more severe on the axle, the correctness of that supposition may fairly be questioned; and it would appear to have been fully contradicted by the lengthened experience gained of the rolling stock of the Great North of Scotland line. With reference to the coupling of four wheels, the writer adopted the four-coupled arrangement for both passenger and goods-engines,—for passenger-engines, because it cannot but be useful to acquire the greatest amount of adhesion-weight by the simple means of a pair of coupling-rods; and for goods-engines, because, in the coupling of a greater number than four wheels, a greater complexity of mechanism must be incurred, and increased resistance and tear and wear ensues, as compared with four-coupled wheels only, which results in a very doubtful gain in economy of locomotive power.

Mr. Cowan has ably studied and matured the engine now under notice, with a view to working on the sinuous line and branches of the "Great North" system, with four-coupled wheels, and a bogie in front. The axles of the bogie are judiciously placed at 6 feet apart between centres, which, as compared with the gauge of the rails, is fully a fourth more, and insures self-adjustment of the bogie to curves, without undue frictional contact with the rails. The radial distance of the front axle from the central pivot —viz., 3 feet—is the leverage through which the outer rail operates in deflecting the bogie, which, it may be assumed, is the least radius that ought to be applied in a bogie on the narrow gauge. The spread of the bogie, moreover, affords sufficient room for the cylinders, which are thus enabled to be placed horizontally between the wheels; and thus the rocking movement, to which engines with inclined cylinders are disposed, more particularly bogie-engines, which take a central bearing on the pivot, is reduced to minimum limits. The bogie is strongly and compactly constructed with plate-framing, in which the usual spider-web diagonals are dispensed with; the necessary horizontal stiffness being amply secured by means of horizontal plates, applied above and below, and uniting the longitudinal and transverse plates of the frame. The longitudinals are made with inside bearings for the axles; and thus, as compared with a bogie having outside frame-plates and axle-bearings, the frame is compressed within a very narrow compass. In Mr. Cowan's engine the bogie-frame measures 3 feet 6 inches wide and 8 feet 4 inches long, covering an area of 29 square feet; whereas, in previous classes of engines with outside-framed bogies and outside bearings, with the same length of wheel-base, the frame is 7 feet wide and 9 feet 6 inches long, covering an area of 66½ square feet, or more than twice the area of the more recent bogie. Of course, the more widely-spread frame must be very much heavier, in order to be as strong as the other. The pivot, which is a part of a sphere 8 inches in diameter, takes its bearing in a cast-iron cup, formed on a sliding block, bearing on steel slides, upon which a lateral traverse of 1 inch is permitted each way from the central position. The block slides between two transverse plates, forming part of the bogie-frame. It is, of course, of the utmost importance that the pivot should be well secured to the engine-frame; and for this purpose it is expanded upwards into a flat web, 3½ inches thick and 9 inches high, inserted between, and firmly rivetted to, two vertical plates, forming part of a strongly-fortified box-framing, just under the smoke-box, but independent of it. For security in case of accident, the pivot is extended down through the sliding block, and is fitted at the lower end with a broad washer and cotter.

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* For an illustration and description of this engine, see *Railway Locomotives*, by D. K. Clark, page 51*.

The frame consists of four longitudinal plates, two on each side, united by six transverse plates on edge, and fortified by horizontal plates, forming the sole of the smoke-box and the platform in front of it. Moreover, the two longitudinal plates on each side are united by the cylinder, which is bolted to them both with flanges and by platform plates and angle-iron. Thus, the whole of the fore part of the frame is strongly bound together into one inflexible piece, so that it may resist both the strain of the bogie and the strain of the cylinders. The inner longitudinal plates are 1 inch thick at the cylinders, and increased to 1¼ inch thick in the after part, where they are 15 inches deep, embracing the driving and hind axles. The outer plates are ¾ inch thick at the cylinders, and reduced to ½ inch thick beyond them; and they are 8 inches deep. The total width of the frame is 7 feet 8 inches, and its length 25 feet 9½ inches. The platform behind the firebox consists of a massive casting, to increase the adhesion-weight.

The cylinders, outside, are lodged between the inner and outer plates of the frame, and are placed at 6 feet 2 inches apart, between centres; they are 1½ inches in thickness. The steam-ports are 1⅜ inches long, by 14½ inches wide, having an area of 20 square inches, or $\frac{1}{10}$ part of the area of the piston. The exhaust-port has twice the area, and the blast-orifice is 4½ inches in diameter, having $\frac{1}{13}$ part of the area of the piston. The pistons are of cast iron, in one piece, with two elastic packing rings, each in one piece. The piston-rods are 2¼ inches in diameter, or $\frac{9}{64}$ part of the diameter of the piston. The connecting-rods are 5 feet 9 inches in length, or 5¾ times the length of the crank. In action they are 1¾ inches thick by 3 inches broad at the neck of the small end, having a sectional area of 5¼ square inches, or 1 inch less than that of the piston-rod. There is one pair of guide-bars, 4½ inches wide, to each cylinder, with slides 16 inches long. The bearing of the cross-head pin is 2¾ inches in diameter by 2¾ inches long, and that of the crank pin is 3½ inches in diameter by 4 inches long, for the connecting-rod; the bearings for the coupling-rods are 3⅜ inches in diameter by 2$\frac{5}{8}$ inches long. The driving axle is 6 inches in diameter, parallel, with bearings 6½ inches in diameter by 7 inches long; it is 7½ inches in the wheel-nave. The hind axle is 6 inches in diameter at the centre, slightly enlarging towards the bearings, which are not cylindrical, like those of the driving axle, but formed to a hollow contour, with a 15-inch radius. The object of the hollow bearing is to take up the end wear of the hind axle, and prevent looseness. The bogie-axles are 5 inches in diameter at the middle, and 5½ inches at the bearing, which is 7½ inches long; they are 6½ inches in the wheels. The driving and hind wheels are made with 18 spokes, spaced at intervals of 10½ inches pitch upon the rim. The tyres are 2½ inches thick, and formed with a ledge let into the outer edge of the rim. The slide-valves are made with 1¼ inches lap, $\frac{1}{32}$ inch lead, and 4$\frac{9}{16}$ inches maximum travel; the eccentrics are 14¾ inches in diameter, and have a stroke of 6½ inches. The eccentric-rods are 4 feet 9 inches long, connected to the expansion-links, which are shifting, at centres 18 inches apart; the valve-spindles are 1⅜ inches in diameter. The motion is in a

THE FOLLOWING IS THE DISTRIBUTION IN FORWARD GEAR:—

Notch.	Travel of Valve.	Lead.		Maximum Opening of Port.		Suppression in Parts of the Stroke.		Release in Parts of the Stroke.	
		Front Port.	Back Port.	Front Port.	Back Port.	Front Stroke.	Back Stroke.	Front Stroke.	Back Stroke.
	Inches.	Inch.	Inch.	Inch.	Inch.	Per Cent.	Per Cent.	Per Cent.	Per Cent.
Full gear, .	4$\frac{9}{16}$ full.	$\frac{1}{16}$	$\frac{1}{32}$	1$\frac{1}{16}$	⅞ full.	65½	67	86½	89½
Third notch, .	4$\frac{1}{16}$ bare.	$\frac{3}{16}$	$\frac{5}{32}$	⅞	$\frac{13}{16}$ bare.	54½	54	80½	84
Second notch,	3$\frac{9}{16}$ $\frac{1}{32}$	$\frac{5}{16}$ full.	$\frac{11}{32}$	$\frac{9}{16}$ $\frac{1}{32}$	⅝	39	37½	72	76½
First notch, .	3¼ full.	⅜	⅜	⅜ full.	⅜	23½	22½	61	66½

direct centre line from the eccentrics to the valve-spindles—a fact favourable for stability and durability. The reversing shaft is 3 inches in diameter, worked by an ordinary lever and notched sector.

The barrel of the boiler is 10 feet 11 inches long, and telescopic, as it is called, having three circles of 7/16-inch plates, successively lapped one within the other, making the diameter next the firebox 4 feet 1 inch, and at the smokebox 3 feet 11¼ inches. The firebox shell is made, after the good old fashion, to a larger radius than the barrel, with a steam-dome 18 inches in diameter on the top; which is a better practice than the "flush-top" boiler, as it gives more steam-room, and a better command of dry steam. The firebox is made of ½-inch copper plates, and measures about 4 feet long at the grate, by about 3 feet 3 inches wide, giving 13 square feet of grate area. The walls of the firebox fall in a little, and give a water-space tapering upwards from 2½ inches at the bottom to 3½ inches at the top, except on the sides, where the space is considerably wider, owing to the circling of the shell to a radius of 2 feet 4 inches. There are 175 flue-tubes, 1¾ inches in diameter, spaced at ¾ inch clear apart, arranged in vertical rows. The sure effect of such good proportions—uniting a moderate number of tubes, but sufficient in heating surface, with plenty of clear space between them for circulation of water and generation of steam—is perceived in the remarkable steaming power of the engine, and is indicated by the significantly liberal proportion of the blast-orifice, the sectional area of which is 1-10th of that of the chimney at the base, and 1-117th part of the area of the firegrate.

The firebox is fitted with Mr. D. K. Clark's smoke-consuming apparatus, which is applied to all the engines on the "Great North" line. On this system, which was patented in 1857, and practised in 1858, air is admitted above the fire through two rows of air-tubes,—one through the front, and the other through the back of the firebox, as shown in detail on Plate XLII.; and a small jet of steam from the boiler is delivered through each air-tube, so as to draw the air-currents into the firebox, and with such precision and velocity as to sweep the surface of the fuel, and forcibly to distribute the air amongst, and thoroughly mix it with, the combustible gases. The air-tubes are opened or closed by means of the sliding damper, as may be required.*

VIII.—Mixed-traffic engine, constructed by Messrs. Fox, Walker, & Co., Bristol, for the Windsor and Annapolis Railway, Nova Scotia. Plate XLVI.

Area of Grate,		16·6 square feet.
Heating Surface,		947·5 ,,
Diameter of Cylinders,		16 inches.
Stroke of Pistons,		22 ,,
Diameter of Driving-wheels,		5 feet 1 inch.
Length of Wheel-base,		20 ,, 3 ,,
Do.	to centre of Bogie,	17 ,, 9 ,,
Do.	of Bogie,	5 ,, 0 ,,

This engine is, in general design, in accordance with the prevailing American type of passenger-engines,—having four coupled-wheels and a bogie. The resemblance ends there; for the engine is entirely English in detail, which appears to be much appreciated by the officials of lines on which the engine is at work. The bogie is strongly constructed with plate-framing and inside-bearings; and besides having the ordinary rotating movement on a pivot, it may traverse laterally for a few inches. A slight lateral movement has also been provided for in the axle-boxes of the driving-wheels. There are 150 flue-tubes, 2 inches in diameter; and as the barrel of the boiler is 4 feet 2 inches in diameter, there is abundant space for the generation of steam and circulation of water. The tender holds 2,000 gallons of water, and is carried on two four-wheel bogies.

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* For a full account of Mr. D. K. Clark's system of smoke-consumption, see the writer's works on *Railway Locomotives*, pages 30*, 36*, and *The Exhibited Machinery of 1862*, page 6; and also a paper "On Coal-burning and Feed-water-heating in Locomotives," read by him at the Institution of Civil Engineers, printed in vol. xix. of the *Proceedings*, Session 1859-60; also Chap. XXX., following.

CHAPTER XXVI.

EXAMPLES OF STEEP-GRADIENT TANK-LOCOMOTIVE ENGINES.

1. Steep-gradient tank-locomotive, with bogie, designed by Mr. John Kershaw, and constructed by Messrs. Sharp, Stewart, & Co., Manchester, for the Great Indian Peninsula Railway. Plate VI.

2. Steep-gradient tank-locomotive, designed and constructed by Mr. John Haswell, Vienna, for the Austrian State Railway. Plate XXI.

3. Steep-gradient tank-locomotive, with W. B. Adam's radial axle-boxes and spring-tyres, designed by Mr. James Cross, St. Helen's Railway. Plate XXIII.

4. Steep-gradient tank-locomotive, designed by M. Vaessen, constructed by the Société St. Leonard, Liège, for the Alar and Santander Railway (Isabella II.), Spain. Plate XXXIX.

I.—Steep-gradient tank-locomotive, by Mr. John Kershaw, for the Great Indian Peninsula Railway. Plate VI.

Area of Grate,		25·9 square feet.
Heating Surface,		1,438 "
Diameter of Cylinders,		20 inches.
Stroke of Pistons,		24 "
Diameter of Driving-wheels,		4 feet 4 inches.
Length of Wheel-base,		19 " 10 "
Do.	to centre of Bogie,	18 " 8 "
Do.	of Bogie,	3 " 2 "
Total Weight in working order,		49 tons.
Do.	on the six coupled-wheels,	37¼ "
Do.	on the Bogie,	11¾ "

This class of engine was designed for working the Bhore and Thul Ghauts inclines of the Great Indian Peninsula Railway. The Bhore Ghaut incline is 15¾ miles long, with an average gradient of 1 in 48; the maximum gradient is 1 in 37, and the sharpest curves are one of 15 chains radius, and two of 20 chains radius. The Thul Ghaut incline is 9½ miles in length, and has an average gradient of 1 in 51·5; the maximum gradient is 1 in 37, and the sharpest curve has 17 chains radius. The firebox is constructed, like Mr. Cudworth's, with two long inclined grates and a longitudinal mid-feather, and gives 26 square feet of grate-area. There are 200 flue-tubes, 2 inches in diameter, and placed at $\frac{11}{16}$-inch clear apart, within the barrel of the boiler, 4 feet 6 inches in diameter, for which, and also for the 20-inch inside cylinders, there is ample room on the 5 feet 6 inch gauge of the Indian Railways. The bogie is not only free to rotate, but also to slide laterally under the engine, to a certain extent. Thus the engine, in passing round curves, turns on the fixed wheel-base, which is only 12 feet 10 inches long, as the bogie is, in virtue of the lateral freedom, placed there rather to carry the leading weight than to guide the engine. The middle driving-wheels are made without flanges, truly cylindrical; and thus the duty of really guiding the engine appears to devolve chiefly upon the leading coupled-wheels, which, with the comparatively short fixed base, may be done without much labour. Indeed, some such latitude was indispensable with such a strongly-built frame, constructed of two longitudinals on each side, bonded together into something absolutely inflexible. It is, however, to be remarked that the bogie is too short: the axles are only 3 feet 2 inches apart, and this excessively short base, in conjunction with the great width of gauge of the rails, 5 feet 6 inches, must be fatal to the free action of the bogie. The construction of the frame of the bogie is commendable: it is compactly built of plates, horizontally and vertically disposed, rivetted together, and with inside bearings,—an arrangement insuring the maximum of strength and stiffness, and the minimum of material. It is to be regretted that the crank-arms of the driving-axle should have been made only 4½ inches thick, when there was room for a greater thickness, more particularly as the cylinders are 20 inches in diameter. True, the crank-arms are made very wide—13 inches in width—which gives a sectional area of 58½ square inches; but the strength of a crank-arm lies in the thickness more than in the width; and a section 5½ inches thick by 10 inches wide, giving an area of 55 square inches, would, if it had been adopted, have given a crank-arm stronger than the actual one in the proportion of 30 to 26. The cylinders, 20 inches in diameter, appear to have been made larger than was useful, with respect to the driving-weight of the engine. Assuming that the effective mean pressure in the cylinders is 100 lbs. per square inch, the total propelling force, as measured at the rails, would be 8¼ tons, or between a fourth and a fifth of the driving-weight (exactly $\frac{1}{4\cdot52}$); and, of course, when the tank, which holds 1,050 gallons, or 4·7 tons of water, is empty, and the coal-boxes exhausted, the driving-weight would be reduced to less than 32 tons, and the cylinder-power of traction would be more than a fourth of the driving-weight (exactly $\frac{1}{3\cdot9}$). Now, calculating even on a propelling force at the rails of one-sixth of the maximum adhesion-weight, which is a large fraction, it would follow that a 17½-inch cylinder would have been sufficiently large in diameter. The engine is constructed with four slide-brakes, on Laignel's system, and upon which the entire driving-weight of the engine may be placed,—thus avoiding the destructive wear of tyres which ensues upon the application of a brake to a heavy engine descending ten to fifteen miles of steep inclines continuously.

II.—Steep-gradient locomotive, by Mr. John Haswell, Vienna, for the Austrian State Railways. Plate XXI.

Area of Grate,	15 square feet.
Heating Surface,	1,323 "
Diameter of Cylinders,	18½ inches.

Stroke of Pistons, ...	24⅝ inches.		
Diameter of Driving-wheels, . .	3 feet	3½ inches.	
Length of Wheel-base, . . .	19 „	8	„
Total Weight of Engine in working order,	46 tons	0	cwts.
Do. at Leading-wheels, .	9 „	3	„
	8 „	5	„
Do. at Middle-wheels, .	9 „	14	„
	6 „	16	„
Do at Trailing-wheels, .	12 „	2	„

This engine, the "Steierdorf," is a tank-locomotive on the Engerth system, with five coupled axles. The boiler rests on two frames coupled together by a pivot-bolt. The steam-cylinders are fixed on the fore part, the tank on the hind part, and the wheels on the fore frame are coupled with those of the hind frame by means of an intermediate shaft and radiating parallel-motion, ingeniously contrived to adapt itself to the varying angularity of the hind frame, on curves, and at the same time to continue to transmit the power from the first to the second frame. This plan of connection has been designed to supersede Engerth's mode of coupling the axles by spur-wheels. The intermediate shaft was proposed in 1852 by Mr. Kirchweger, and the application, as here shown, was worked out by Mr. Pius Fink. The intermediate shaft has overhung cranks, and is carried above and by the fore axle of the tender, on spherical bearings; and it is connected to, and at a proper distance apart from, the hind axle of the engine, by rods, also by spherical bearings. The relative positions of the three axles being thus fixed, the power is transmitted from the hind or driving axle of the engine, by coupling-rods, to intermediate cranks, and thence to the outside cranks of the fore tender axle, and further by coupling-rods to the hind axle. The wheels are 3 feet 3½ inches diameter, the cylinders 18½ inches by 24⅝ inches. The total weight in working order is 46 tons. The firebox, made for burning coke, has 15 square feet of grate; and with great good sense and reticence, there are introduced only 158 flue-tubes, $2\frac{1}{11}$ inches diameter, with plenty of clear space between them.

III.—Steep-gradient tank-locomotive, by Mr. James Cross, St. Helen's Railway. Plate XXIII.

Area of Grate, ...	16·25 square feet.		
Heating Surface, ...	687 „		
Diameter of Cylinders, ...	15 inches.		
Stroke of Pistons, ...	20 „		
Diameter of Driving-wheels, . . .	5 feet 1 inch.		
Length of Wheel-base, total, . . .	22 feet.		
Do. rigid, . . .	8 „		
Total Weight of Engine in working order,	40 tons	15	cwts.
Weight on Leading-wheels, . . .	7 „	15	„
Do. Driving-wheels, . . .	11 „	15	„
Do. Coupled-wheels, . . .	11 „	5	„
Driving-weight, ...	23 „	0	„
Weight on Hind-wheels, empty, . .	4 „	10	„
Water, ...	4 „	5	„
Fuel, ...	1 „	5	„
Weight on Hind-wheels, loaded, . .	10 „	0	„

This engine was designed to avoid the acknowledged evils of long rigid wheel-bases, particularly for the ascent of steep inclines in combination with sharp curves; and for this object it was constructed with the radial axle-boxes and spring-tyres invented by Mr. W. B. Adams. The driving and coupled axles are placed respectively in front and behind the firebox, 8 feet apart, and the leading and trailing axles are placed each at 7 feet from the coupled axles, or at an extreme distance of 22 feet apart, and are fitted with radial axle-boxes, which are formed as segments of circles, and are free to traverse laterally and radially between the guides fixed on the frame. Krupp's tyres are applied to all the wheels, with springs between the tyre and the rim, as shown in the annexed cut, Fig. 217.

The firegrate is 5 feet long; there are 121 flue-tubes, 10 feet 11 inches long, 1¾ inches outside diameter, and ⅝ inch clear—a disposition worthy of commendation, because it provides for free circulation and generation of steam. The pressure in the boiler is 140 lbs. per square inch. The tank holds 950 gallons of water, and the bunker 25 cwts. of coal. The following additional particulars of this engine are derived from a paper read by Mr. Cross, at the Institution of Civil Engineers, in April, 1864:*—

"The novelty of the engine is in the radial axle-boxes, which allow an engine on one rigid frame, carrying both boiler and tender, to pass round extremely sharp curves with an easy gliding motion, while the great length controlling all irregularities, makes the engine very steady at high speeds. The construction in effect reduces the real wheel-base of 22 feet to 8 feet when passing round curves, this being the distance between the coupled-wheels.

"The radial axle-boxes are only a little larger than ordinary engine axle-boxes; but, instead of being square to the framing, are struck with a radius which has its centre in the centre of the adjoining axle, giving in this case a radius of 7 feet, which the axle-box guide-blocks are curved to fit. The boxes are allowed to play laterally 4½ inches on each side; and the spring-pins, instead of being fixed immediately on the top of the boxes, are each fitted with a small slide or roller, so as to allow the boxes to traverse freely. It will be at once seen that the latter, though of a different shape, are still only axle-boxes weighing 3½ cwts. each, and that the only additional parts added to the engine are the spring-pin rollers. This simplicity so completely answers the desired end, that it is claimed as one of the best features of the new engines. The action of the axle-boxes is as follows. On entering a right-hand curve the flanges of the leading wheels draw the boxes to the right, the framing remaining as a tangent to the curve; and, as the axle-boxes are themselves curved, the result is to bring the axle on the right hand side nearer to the driving-wheel, and to push it on the left-hand side further from the driving-wheel. The trailing axle, in its turn, takes a similar position, and the leading and trailing axles thus become radii of the curves they are traversing, and keep their flanges always parallel to the rails; and the boxes, moving freely from right to left, adjust themselves to the multiplicity of short S curves caused by irregularities in the rails. By looking over the side while the engine is moving at high speeds, the boxes may be observed to move with an incessant lateral vibration, while the frame remains steady (thus showing the number and amount of blows which the frame and rails receive in ordinary cases), the friction, in passing round curves, being limited to the coupled-wheels or 8-feet base. This 8-feet rigid base is, however, again greatly eased in passing round curves, by the use of the spring-tyres, which admit of a slight lateral motion. These tyres also give a better grip on the rail, by slightly flattening under the weight of the engine, and thus presenting more surface for friction. As a proof of this, a train of waggons was placed on a curve of 450 feet radius, with a rise of 1 in 78, and two engines, identical in every respect (except that the wheels of one engine had spring-tyres), worked at the same pressure, and driven by the same man, were successively attached to it, when the one with spring-tyres brought out 15 waggons, and the one with ordinary tyres 13 waggons. This result is confirmed by the working of all the engines fitted in the same way.

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* *Minutes of Proceedings of the Institution of Civil Engineers*, vol. xxiii., 1863-64.

"This engine was completed in the first week of November, 1863, and has since been running regularly, taking its turn of duty with passenger trains, or coal trains, or as a shunting engine, and about the numerous works connected by sharp curves with the St. Helen's line. The motion round curves is free from all jerking, and on straight lines the speed is more than sixty miles an hour, either end of the engine being first, without any train behind to give steadiness; and the motion is so smooth, that it has only been by taking the actual time, that engineers have convinced themselves of the fact of the speed exceeding forty miles an hour. It was built to traverse curves of 200 feet radius. This it does with the greatest facility, and it has regularly worked the passenger trains round a curve of 1,000 feet radius, going directly off the straight line by a pair of facing points, at a speed of more than thirty miles an hour; and it has gone round a curve of 132 feet radius. It has also run a train of twelve passenger carriages, weighted up to 100 tons, exclusive of its own weight, at sixty miles an hour on the level. From the advantages it possesses over ordinary mixed engines for weighting the trailing coupled-wheel, it, without difficulty on a wet slippery day, started and took this load up a gradient of 1 in 70, drawing seven of the carriages with a load weighing 72 tons 5 cwts., up a gradient of 1 in 36, round a curve of 440 feet radius; and coal trains of 250 tons are worked over long gradients of 1 in 200 with the greatest ease. It is evident, then, that engines on this principle, affording facilities for the use of high power in hilly countries, are peculiarly adapted for metropolitan lines, where sharp curves are a necessity (being equally safe whichever end is foremost), and are also well suited for light lines in India and the Colonies. It may likewise be remarked that carriages and waggons on this principle would carry heavier freights, with a saving in the proportion of dead weight, while their friction round curves would be less than at present."

With regard to the performance of the spring-tyres with which the engine is fitted, Mr. Adams, the inventor, communicated some details of their performance in a separate paper, read at the same time,[*] written with his usual clearness of perception and force of expression, from which the following is derived:—

"A loose tyre is one of the things denounced as a source of danger on railways, and to be carefully guarded against. It is most dangerous in the case of a tyre fastened with through-bolts or rivets, because it indicates that the tyre has lost the true condition of a true circle, and has become a ring of many segments, constantly striking blows. But a tight tyre is also a source of danger, by reason of its tendency to burst, especially in time of frost. The author, therefore, came to the conclusion that a tyre neither loose nor tight, but fitting elastically to the wheel, was the safest of all, as there can be no tendency to burst, with sufficient yield to prevent the mischievous effect of blows, as a result of keeping constantly in contact; while a slightly-pliant surface induces a better adhesion of the driving-wheels, and freer movement of the train-wheels. There is an apparent objection to change of form in the circle of the tyre, but not if the tyre be elastic.

"Being satisfied of these conditions, the author devised the following plan. A tyre, with a deep front rib, was bored out internally to a cone, parallel to the tread of the tyre: this cone was hollowed out to a depth of a quarter of an inch, leaving a flat edge of three-eighths of an inch on each side. On this was laid a hoop of spring steel, tempered. The periphery of the wheel was then turned to a curve across the breadth, corresponding to the curve in the tyre. Into the cone formed by the spring the wheel was pressed, with a force equal to three bolts of 1 inch in diameter, and confined there by a flat ring, sprung into the groove in the back of the tyre. The author was satisfied with the principle and the mechanical arrangement; but there remained the great difficulty of persuading engineers of the advantages of the system. At length an opportunity for making the experiment was conceded. A set of wheels was selected of the class known to be the most destructive to tyres—solid discs of wrought iron—which, by their rigid action, crushed the metal on the rails. The peripheries of the wheels were turned to a curve; the hoop-spring below was $\frac{1}{6}$th of an inch thick, and 3½ inches wide. The tyres were of Staffordshire iron, with a deep front rib, and a false rib or flat ring in a groove at the back. The wheel was forced in on the steel hoop by the pressure of two bolts, each 1 inch in diameter, and the flat ring was sprung into its

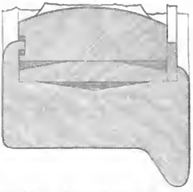

Fig. 217.—Section of Engine spring-tyre, showing a steel band interposed between the wheel and the tyre.

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[*] *Minutes of Proceedings of the Institution of Civil Engineers*, vol. xxiii., 1863-64.

place with elastic expansion. This set of wheels was applied to a heavy carriage on the North London line, and ran upwards of 106,000 miles without being turned up, and with scarcely any perceptible difference in their surface, the flanges being absolutely unworn; while Low Moor tyres, under the same conditions, were heavily worn, both in flange and tread.

"On another line, the four coupled-wheels of a six-wheel engine were fitted with similar tyres, which ran about twelve months. These were very imperfect. The tyres of Staffordshire iron, badly made, were reedy, and separated into strips. The springs were in single layers, were too weak for the work, and had to be replaced; but some remarkable facts were developed. The tyres, when so loose as to be moved round the lifted wheel by hand—in fact, having ¼ inch play—were found equally efficient adhesion and draught. The tyres did not slip on the rails, but the wheels did slip within the tyres, just sufficiently to compensate for the difference in length of pathway round sharp curves. Finally, however, this engine was broken up, the boiler being too old to be used with safety.

"On the same line, on curves where the wear on the leading-wheels of the engine was so severe, that with new wheels, after running only 9,000 miles, the flanges needed 'turning up,' a pair of Leeds tyres was applied with springs, and ran 25,240 miles before 'turning up' was required. At the same time a similar engine was contrived with a 'bogie' for the leading axle to transverse upon curves, so as to give abundant lateral traverse to the wheel cones, with the same quality of tyres; and the result was, that the spring-tyres, without this advantage, gave an equally good result.

"This new system of wheels, but with double instead of single spring hoops, was applied by Mr. Cross, the engineer of the St. Helen's Railway, Lancashire. This line is one of the most remarkable for its heavy gradients and sharp curves, the gradients being 1 in 35, 1 in 70, and 1 in 85, while the curves are of 300 feet and 500 feet radius; and, moreover, the points and crossings are very frequent, twelve miles of sidings occurring in a distance of two miles from the St. Helen's station, the total length of the line being thirty miles.

"Mr. Cross states date September 4, 1861, that 'the tyres are giving great satisfaction. I had them made of common Staffordshire iron, and put under a six-wheeled coupled engine, weighing 21 tons, and they have now run 29,429 miles with heavy trains. On this line the gradients are all heavy, with the sand constantly in use, yet the wear has been very slight. I am so satisfied, that when I get a new locomotive, I shall have similar tyres.'

"From the annexed tabular statement of the performance of four classes of tyres on the St. Helen's Railway, compared with each other—viz., Krupp's steel, Swedish iron, Hood & Cooper's best Yorkshire iron, and Staffordshire iron—with springs between the tyre and the wheels, it appears that Swedish iron (engine No. 27) has less than three-fourths the durability of Hood & Cooper's Yorkshire; that Krupp's steel has run twice the distance of Hood & Cooper's; and comparing Staffordshire tyres with springs (engine No. 18) with Krupps steel (engine No. 23), it appears that the former has two more coupled-wheels one-ninth less in diameter, one-thirteenth more weight, has run one-third more mileage, scarcely ever slips, and that the tyres have not been turned up, or needed it. The total distance run by the Staffordshire tyres, before reducing the flanges, was 63,913 miles, as against Krupp's 40,843 miles. But, inasmuch as Krupp's steel tyres were 4 feet 6 inches in diameter, and the spring-tyres, were only 4 feet in diameter, credit must be taken for one-eighth more revolutions in the latter, making the distance run by the latter equivalent to 71,902 miles, as against 40,843 miles of the former."

COMPARATIVE WEAR OF FOUR CLASSES OF ENGINE TYRES ON THE ST. HELEN'S RAILWAY, LANCASHIRE.

No. of Engine.	No. of Wheels.	No. of Coupled.	Diameters of Wheels.	Weight.	Miles Run.	Slip of Wheels.	Remarks.	Class of Tyre.
			Ft. In.	Tons. Cwts.				
23	6	4	4 6	19 15	40,843	Slipped, even in dry weather.	Needing turning up.	Krupp's Steel.
4	6	4	4 6	20 6	20,798		Needing turning up.	Hood & Cooper's best Iron.
7	6	4	4 6	23 14	34,006		Needing turning up, and been turned once.	Swedish Iron.
18	6	6	4 0	21 0	55,138	Scarcely ever slip.	Never turned up, nor needing it.	Staffordshire Iron, with springs between the wheel and the tyre.
					71,902	......	Before being turned up.	

The sections, Figs. 218, 219, and 220, show in dot lines the wear of three out of these four different classes of engine tyres; all the templates of the wear being taken from plaster casts.

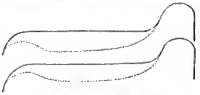


Fig. 218.—Engine No. 23.

Wear of Krupp's steel tyres, 4 feet 6 inches diameter. The upper section is from the driving tyres; load on pair, 9 tons 18 cwts. The lower section is from the leading tyres; load on pair, 7 tons 17 cwts.

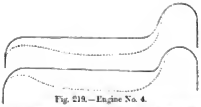


Fig. 219.—Engine No. 4.

Wear of Hood & Cooper's best Leeds iron tyres, 4 feet 6 inches in diameter. The upper section is from the leading tyres; load on pair, 6 tons 10 cwts. The lower section is from the driving tyres; load on pair, 9 tons.

"It will be perceived that, with the exception of reducing the depth of the flanges, the spring-tyres do not need turning up, and apparently may be worn out without turning up, save the reduction of the flanges.

"That wear occurs at all is, however, an indication that yet more is required in principle. It should be observed, that the tyres most worn on the six-wheeled coupled engine, are those with the smallest load. The reason for this apparent paradox is, that the load on the leading wheels is probably not enough to give adhesion on the rail, and to force the wheels to slip within the tyres for compensation, which the more heavily-loaded wheels do.

"The principle needed, in addition to the moving tyre, is that of radiating axles. The flanges of the spring-wheels are saved from wear on curves, partly by slipping round, and partly by horizontal rocking on the curved periphery of the wheel. The flange of a rigid tyre acts as a sheering machine; it cuts the rails, and is cut by the rail. The flange of a spring-wheel, if it touches the rail, yields and springs up, while the tread fits itself to all inequalities of the rail. But the spring-tyre, in common with the rigid tyre, is occasionally obliged to slide across the rail-table laterally, and this action is mischievous, though in a less degree."

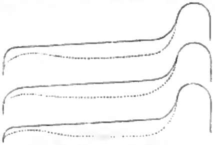


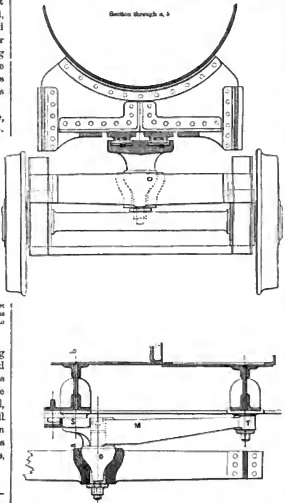
Fig. 220.—Engine No. 18.

Wear of Staffordshire iron tyres, with springs between the tyre and the wheel, 4 feet in diameter. The uppermost section is from the trailing tyres; load on pair, 5 tons 6 cwts. The middle section is from the driving tyres; load on pair, 7 tons 14 cwts. The lowest section is from the coupled driving tyres; load on pair, 7 tons 16 cwts.

IV.—Steep-gradient passenger tank-locomotive—M. Vaessen's system—constructed by the "Société St. Leonard," Liège, for the Alar and Santander Railway, Spain. Plate XXXIX.

Area of Grate, ...	27·5 square feet.
Heating Surface, ...	1,261 "
Diameter of Cylinders, . . .	18·1 inches.
Stroke of Pistons, ...	24 "
Diameter of Driving-wheels, . .	5 feet 6⅛ inches.
Length of Wheel-base, extreme, .	18 " 4½ "
Do. to centre of Bogie,	16 " 1 "
Do. fixed, . .	7 " 6½ "
Do. of Bogie, . .	4 " 7 "
Total Weight in working order, with full supply of fuel and water, .	45 tons.
Mean Weight in working order, .	42 "
Do. on coupled-wheels, .	25 "
Do. on Bogie wheels, .	17 "

This engine is placed on two pairs of coupled-wheels, and a four-wheel bogie in front. The bogie is articulated in a peculiar manner, on M. Vaessen's system, for the same purpose as the system of Mr. Bissell, to admit of lateral play on curves. (See Figs. 221 and 222.) The truck radiates on a pivot, T, fixed to framing under the boiler, by means of a radius-lever, M, the other end of which embraces, and is fixed to, the central spherical pivot, O, which supports the weight of the fore end of the engine, and rests and turns in a socket prepared for it in the frame of the bogie. Sliding surfaces, S, S, are fitted to the lever, to carry the weight, forming inclined planes, which admit of lateral play of the lever in curved portions of the line, and at the same time tend to restore the lever to the central line of the engine. In the plan, Plate XXXIX., the engine is represented standing on a curve of 656 feet radius, showing a displacement of the pivot equal to 1⅜ inches. It is apparent, however, that the spherical pivot is an unnecessary addition to the radius lever, and needlessly adds to the complexity and cost of the work, as is proved in Mr. Fowler's engine, on the Metropolitan Railway, Plate



Figs. 221 and 222.—Sections of Bogie, on Vaessen's system. T, pivot; M, lever; O, spherical central pivot of bogie; S, S, sliding surfaces.

XXXVI., to be described in the following chapter, which has only a radius lever fixed to the bogie-frame. The fuel-boxes and water-tanks are placed on the two sides of the engine, over the coupled-wheels, and thus the dead weight on the bogie is relieved. The firebox is 7 feet 10½ inches long, with an inclined grate, on Belpaire's system, having an area of 27½ square feet, for burning coal-slack. The barrel of the boiler is 4 feet 3 inches in diameter, made of ½ inch plates, with 200 flue-tubes, 2 inches in diameter, and 11 feet 2½ inches in length. The working pressure in the boiler is 7 atmospheres, or 103 lbs. per square inch; the water-tanks hold, when full, about 4½ tons of water, or 1,000 gallons; and the coal-boxes hold 1½ tons of slack. The valve-gear employed is Walschäert's gear, the principle of which has already been described (page 206).

The lateral displacement of the bogie is calculated for different curves, as follows:—

250 mètres	(820 feet),	. .	28 millimètres	(1⅛ inch).
200 "	(656 "),	. .	35 "	(1⅜ ").
150 "	(492 "),	. .	46·6 "	(1⅞ ").
100 "	(328 "),	. .	70 "	(2¾ ").

Three engines of this class were delivered in 1863.

A number of goods tank-locomotives, on the same system, with six coupled-wheels, and a bogie in front, were constructed in 1861, by the St. Leonard Company, for the same railway. From the elevation, Fig. 223, it appears that the goods engines are similar to the passenger engines, and the leading particulars are subjoined.

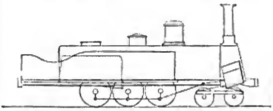


Fig. 223.—Goods Tank-locomotive, for Steep Gradients—Vaessen's system.

Steep-gradient Goods Tank-locomotive (Vaessen's system):—

Area of Grate, ...	27·5 square feet.
Heating Surface, ...	1,502 "
Diameter of Cylinders, . . .	18·1 inches.
Stroke of Pistons, ...	24 "
Diameter of Driving-wheels, . .	4 feet 1¼ inches.
Length of Wheel-base, extreme, .	18 " 4 "
Do. to centre of Bogie, .	16 " 4 "
Do. fixed, . .	8 " 6 "
Do. of Bogie, .	4 " 0 "
Total Length over the Buffers, . .	35 " 0 "
Weight of the Engine, empty, . .	36 tons.
Do. do. full of water and fuel, .	45 "
Do. on the six coupled-wheels, .	36 "
Do. on the Bogie, ...	9 "

The firebox is of the same length and breadth as that of the passenger engine, but is higher. The tubes, also, are the same in number and diameter, but are 13 feet 6 inches long; hence chiefly is derived the excess of heating surface of the goods engines over that of the passenger engines.

The lateral displacement of the bogie on curves is the same as has already been exemplified for that of the passenger engine, the length of wheel-base being substantially the same. It may be added that for the curves of given radius, the lateral play of the middle coupled axle, in the axle-boxes, should be as follows:—

300 mètres radius	(984 feet),	. . .	·114 inch.
250 " "	(820 "),	. . .	·132 "
200 " "	(656 "),	. . .	·172 "
150 " "	(492 "),	. . .	·228 "
100 " "	(328 "),	. . .	·340 "

The performance of these excellently-designed goods engines has been investigated and reported by M. C. G. de la Vega, engineer-in-chief of the railway (Isabella II.); and the following particulars are derived from his report, dated January, 1863, which appears to have been faithfully and intelligently prepared. The first engine, No. 16, started in June, 1861, and the other seven engines followed in succession. From that date, to the end of June, 1862, they had run 122,367 miles, having drawn

PERFORMANCE OF GOODS TANK-LOCOMOTIVES (VAESSEN'S SYSTEM) ON THE ALAR AND SANTANDER RAILWAY.

		No. 16.	No. 17.	No. 20.
Miles run, ...	Miles,	71·3	74·4	74·4
Maximum gradient, ascending, ...	.	1 in 50	1 in 50	1 in 50
Do. descending, ...	.	1 in 100	1 in 100	1 in 100
Number of waggons in train, ascending, ...	.	22	21	17
Weight of train, ...	Tons,	177	190	145
Number of waggons in train, descending, ...	.	27	24	24
Weight of train, ...	Tons,	340	300	300
Average speed in miles per hour, ...	.	12·4	12·4	12·4
Coal consumed, ...	Lbs.,	3,520	3,511	3,630
Do. per mile, ...	"	49·4	47·0	48·8
Water evaporated, ...	"	30,668	28,985	34,009
Do. per mile, ...	"	430	390	457
Do. per lb. of coal, ...	"	8·71	8·26	9·36
Steam pressure in the boiler, ascending, lbs. per square inch, .	.	85 to 115 lbs.	85 to 115 lbs.	90 to 115 lbs.
Do. do. descending, do., .	.	75 to 95 lbs.	65 to 95 lbs.	75 to 95 lbs.

more than 450,000 tons, or, approximately, having drawn 26,660,000 tons one mile. In running these 122,367 miles, a total of 3,069 tons of coal-slack were consumed, being at the rate of 56 lbs. per mile. It was expected that, under favourable conditions, the consumption of fuel would not exceed 45 lbs. per mile. In the trip from Santander to Barcena, thirty-four miles long, on an ascending gradient, of which the maximum incline is 1 in 50, the consumption of fuel is 74 lbs. per mile; returning, it is only at the rate of 20 lbs. per mile. These engines, according to the terms of the contract, were required to draw a train of 200 tons weight, up an incline of 1 in 50, at the rate of from twelve to thirteen miles per hour—a condition which has been easily fulfilled. The usual weight of train is from 170 to 180 tons weight. On an ascending gradient of 1 in 100, trains of from twenty-five to thirty loaded waggons are drawn, or a total train-weight of about 400 tons.

The preceding table contains results of performance of three engines of this class, with six ordinary trains, making three double trips between Barcena and Santander.

CHAPTER XXVII.

EXAMPLES OF TANK-LOCOMOTIVES FOR ORDINARY RAILWAY TRAFFIC.

1. Mineral Tank-locomotive, designed and constructed by Messrs. Fletcher, Jennings, & Co., Whitehaven. Plate XXII.

2. Passenger Tank-locomotive, designed by Mr. John Fowler, C.E., London, constructed by Messrs. Beyer, Peacock, & Co., Manchester, for the Metropolitan Railway. Plate XXXVI.

3. Passenger Tank-locomotive, designed and constructed by Mr. William Adams, Locomotive Superintendent, Bow, for the North London Railway. Plates XXXVII., XXXVIII.

4. Goods Tank-locomotive, designed and constructed by Messrs. Manning, Wardle, & Co., Leeds. Plates XLIII., XLIV., XLV.

5. Mixed-traffic Tank-locomotive (on Fairlie's system), designed and constructed by Mr. A. M'Donnell, Locomotive Superintendent, for the Great Southern and Western Railway of Ireland. Plate XLVII.

I.—Mineral Tank-locomotive, by Messrs. Fletcher, Jennings, & Co., Whitehaven. Plate XXII.

Area of Grate, ...	5·77 square feet.
Heating Surface, ...	426 "
Diameter of Cylinders, . . .	10 inches.
Stroke of Pistons, ...	20 "
Diameter of Driving-wheels, . .	3 feet 4 inches.
Length of Wheel-base, . . .	6 " 0 "

This is a small engine suitable for collieries and quarries, and low speeds. It is fitted with Allan's straight-link motion, which was probably the most convenient of application, as the whole motion is sustained by the reversing shaft. The link-motion is also for convenience worked off the leading axle, which is coupled to the driver—an arrangement which admits of the driving axle being placed under the firebox. But the movement is transmitted with some degree of obliquity from the expansion-links to the valve-spindles, and there is no doubt that such obliquity of action must shake the freely hung pendulous gearing considerably at high speeds.

II.—Passenger Tank-locomotive, designed by Mr. John Fowler, C.E., London, constructed by Messrs. Beyer, Peacock, & Co., Manchester, for the Metropolitan Railway. Plate XXXVI.

Area of Grate, ...	19 square feet.
Heating Surface, ...	1,014 "
Diameter of Cylinders, . . .	17 inches.
Stroke of Pistons, ...	24 "
Diameter of Driving-wheels, . .	5 feet 9 inches.
Length of Wheel-base, . . .	20 " 9 "
Do. to centre of Bogie,	18 " 9 "
Do. of Bogie, . .	4 " 0 "

This engine is constructed with four coupled-wheels and a bogie in front, and is particularly remarkable for the condensing arrangements with which it is supplemented, by means of which the exhaust steam, instead of being, in the usual manner, discharged by the chimney, is diverted into tanks of cold water, where it is condensed, during the passage of the engine through the subterranean portions of the railway. At the same time, the ashpan-dampers are closed, and thus the emission of the products of combustion into the atmosphere of the tunnels is almost entirely prevented. The firegrate has 19 square feet of area. The barrel of the boiler, though 4 feet in diameter, holds only 166 flue-tubes, 2 inches in diameter,—which is in accordance with the most intelligent practice, avoiding a vain and hurtful accumulation of tubes and heating surface, nominally so called. By thus limiting the tubes to that moderate number, Mr. Fowler has left abundant room for circulation within the boiler, and for the easy generation of steam for which the engines of the class illustrated are remarkable. The bogie is strongly and compactly constructed, with plate-framing and inside bearings, at 4 feet centres, traversing on the Bissell principle, on a central pin fixed to the framing of the engine, at a radial distance of 6 feet 8 inches from the centre of the truck.

This radial length, 6 feet 8 inches, insures a nearly correct radiality of the bogie to curves of all radii. The proper length of radius to insure exact radiality of the centre of the bogie for all curves would be 7 feet 2 inches, or 6 inches more than the actual length,—a difference which is perhaps of no great importance in practice. The general design and arrangement of the bogie in the Metropolitan engines is probably the best practical application of the bogie that can be made; that is to say, for the purpose of carrying and guiding the engine along straight lines and curves of various radii of curvature.

The framing of the engine is simple, having a single longitudinal plate on each side, with the axlebox-guides forged on solid, for the coupled axles; the longitudinals are united by five transverse plates.

The carriages used on the Metropolitan Railway are very large, the first class holding forty-eight, and the second and third class eighty passengers each. They weigh, when empty, about 15 tons. The trains worked by the engines above described consist of from three to five such carriages, and with this load the engines start readily up inclines of 1 in 100. The normal pressure in the boiler is 130 lbs. per square inch, but this pressure sometimes falls considerably during the passage of the engines through the tunnels, when the dampers are closed and the exhaust steam condensed. From indicator-diagrams which have been taken from the cylinders of these engines whilst the condensers were in use, it appears that there was no exhaust back pressure on the pistons. In order that the water in the tanks may be rapidly discharged when it becomes heated by the exhaust steam, each tank is fitted with a 7-inch pipe leading to a cast-iron valve-box beneath the foot-plate, containing a valve 10 inches in diameter, worked by a screw from the foot-plate, through which the contents of the tanks can be discharged into a pit beneath the engine in about one minute. The exhaust steam is discharged from a 7-inch pipe upon the surface of the water in the tank, and into the mouth of which pipe a smaller pipe, 4 inches in diameter, is introduced a little way, the pipe descending also into the water; thus, the impulse of each blast is communicated to the water, so as to circulate it sufficiently to prevent the heated water from resting on the surface.

III.—Passenger Tank-locomotive, designed by Mr. William Adams, Locomotive Superintendent for the North London Railway. Plates XXXVII., XXXVIII.

Area of Grate,	square feet.
Heating Surface,	1,015 „
Diameter of Cylinders,	17 inches.
Stroke of Pistons,	24 „
Diameter of Driving-wheels,	5 feet 6 inches.
Length of Wheel-base, extreme,	
Do. to centre of Bogie,	
Do. of Bogie,	
Weight of Engine in working order,	43 tons 12 cwts.
Do. on Bogie-wheels,	14 „ 14½ „
Do. on Driving-wheels,	14 „ 5 „
Do. on Trailing-wheels,	14 „ 12½ „
Do. of Engine, empty,	38 „ 12 „
Do. on Bogie-wheels, empty,	15 „ 14 „
Do. on Driving-wheels,	11 „ 11 „
Do. on Trailing-wheels,	11 „ 7 „

This engine was turned out in 1868, and is the first of the standard class of outside-cylinder engines for passenger traffic, designed and constructed by Mr. William Adams, for the North London Railway. It is made with four coupled-wheels, 5 feet 6 inches in diameter, with an adhesion-weight of nearly 29 tons. The bogie, having 2 feet 9 inch wheels, carries 14 tons 14½ cwts., when the engine has its load of fuel and water; and, curious to say, the weight on the bogie is a ton more when the engine is empty than when loaded. The pressure in the boiler is 160 lbs. per square inch; and to meet this pressure, the barrel of the boiler is made of ½-inch plates, the inside diameter being 4 feet 1 inch. There are 200 flue-tubes, 1¾ inches in diameter, and judiciously placed ¾ inch clear of each other. Thus, ample space for the generation and circulation of steam is provided.

The cylinders are placed at a very low inclination, the bogie-wheels being spread apart, at 5 feet 8 inch centres, and admitting of the cylinders and their connections being partially depressed between them. The method adopted for fixing the cylinders is new. The valve-chests are passed through, and well-fitted to openings in the frames, and each valve-chest has on the upper and lower side a shallow longitudinal rib, the two ribs being planed to fit corresponding recesses in two castings which extend across the engine from one valve-chest to the other. The two castings—the one above and the other below the valve-chests—are flanged all round, and the small spaces between their side flanges and the frame-plates are filled in with slightly tapered wedge-pieces. By this arrangement it is insured that the two cylinders are exactly parallel, and are maintained at precisely the proper distance apart, it being impossible, when the recesses in the castings and ribs on the valve-chests are correctly planed—and they are all planed to templates—that any error should occur. The castings also form a good stay to the frames at the leading end, and the lower casting, moreover, serves to form a convenient connection for the bogie. The piston-rod guide-bars are double, and each crosshead with its pair of blocks is a single steel casting. The connecting-rods have no brasses at the small ends, but are merely fitted with a steel bush, upon which the rod-end is shrunk. Mr. Adams, like Mr. Ramsbottom, has long used solid bushes in place of brasses for his coupling-rods. The rubbing pieces for the driving and trailing axle-boxes are of cast-steel, and are united with a very broad flange at the top, so as to add materially to the strength of the frames. They are fitted with adjusting wedges. As in all Mr. Adams's engines, the wearing surfaces are throughout very large.

The cast-iron bearing plate, through which the engine rests on the bogie, is stiffened with flanges on its lower side, which are expanded into a circular bearing surface, between which and the frame of the bogie a massive ring of india-rubber is interposed, to carry the weight. This ring is confined between two discs of cast-iron with ledges, the upper one of which is secured to the bearing plate, and is formed with a pivot which passes through the india-rubber ring and descends into an eye formed in the lower disc,—this disc resting on a cast-iron sliding-block, which is capable of lateral transverse movement to the extent of 2 inches each way from the centre line between two transverse members of the bogie-frame. The whole is loosely connected together by a central bolt which passes down through the pivot, with a washer-plate and nut below. With such means of universal freedom of movement, the bogie is enabled to adapt itself to any curvature or inequality of the road. The sliding block, it should be added, traverses the bogie on level surfaces, and is in this respect distinguished from the slides on the Bissell system, which ascend inclined planes in moving laterally; but the lateral movement is kept in check by auxiliary india-rubber springs which are opposed to it, and by means of which the bogie is restored to the central line of the engine when it arrives on a straight portion of the road.

The bogie is constructed of iron plate, with inside bearings, and a bearing-spring on each side, suspended between two flitch-plates, forming a compensating beam, and spanning the two axle-bearings. At the bearings of the coupling-bolts, by which the bogie-frame is suspended from the springs, three thin discs of india-rubber are inserted, above the washer-plates, to soften the contact.

The bearing-springs of the coupled axles are underhung with india-rubber bearing discs and compensating beams.

The standard class of outside-cylinder engines perform a duty of a very heavy nature, in working the passenger trains of the North London line. The length of the line is 16 miles 27 chains, upon which the engines with express trains make six stoppages, at an average interval of 2½ miles, and perform the trip at a speed averaging 27·31 miles per hour. With ordinary trains they stop seventeen times, at intervals of less than a mile, and nevertheless make an average speed of 19·22 miles per hour. The trains consist of twelve carriages, weighing, empty, 90 tons 14 cwts.; and loaded, 112 tons 6 cwts. The average consumption of coal on both services is 30·28 lbs. per train-mile; and 9·001 lbs. of water is evaporated per lb. of coal. The india-rubber springs of the bogie have been (1870), in some instances, in constant work for four years, and have not hitherto shown any sign of wear or injury. They are made of ordinary spring india-rubber; with two coils of steel wire cast within them, which add considerably to their strength. The compression of the bearing-spring is about $\frac{5}{16}$ inch under a load of 10 tons. It is remarkable that the load on the bogie is greater when the engine is empty than when full; this is accounted for by the fact, that the weight of the water in the tanks, and the coal in the bunker, is chiefly behind the driving-wheels.

Taken altogether, this engine, carefully thought out and matured in all its details, and well answering the purpose for which it was designed, is a study for engineers.

IV.—Goods tank-locomotive, by Messrs. Manning, Wardle, & Co., Leeds, for the Great Northern Railway. Plates XLIII., XLIV., XLV.

Area of Grate, ...	10¾ square feet.
Heating Surface, ...	782½ ,,
Diameter of Cylinders, ...	15 inches.
Stroke of Pistons, ...	22 ,,
Diameter of Driving-wheels, . .	4 feet 2 inches.
Length of Wheel-base, . .	15 ,, 3 ,,
Weight of Engine, in working order, .	27 tons 0 cwts.
Do. empty, . . .	22 ,, 10 ,,

This engine has six coupled-wheels, with inside bearings, and a pair of single longitudinal frame-plates, from beam to beam. It has a pair of large tanks, one on each side of the boiler, and a third tank below the foot-plate, holding altogether 830 gallons of water. There is a considerable length of overhang behind the trailing-axle,—considerable, not on account of any excessive weight there, but because of the great distance to which the draw-hook or link is removed from the centre of the engine. From the driving-axle to the hinge-pin of the draw-link the distance is 13 feet 4 inches; from the back of the firebox it is 7 feet 4 inches. The interposition of the auxiliary tank below the foot-plate, of course, has prevented the employment of the customary method of applying the traction-bar—viz., to a draw-bolt close behind or near to the firebox, which is found to be the best situation for it, in facilitating the traction, and making the most of the power of the engine. In fact, the nearer the centre of traction can be, and is placed to the centre of effort, which in the present engine is at or near to the crank axle, the more efficient is the engine for traction. The engine is generally well worked out in its details; the cylinders are solidly united to each other and to the frame-plates; the valve-gearing is very directly connected; the longitudinal frame-plates are well bound together; and the boiler is well proportioned. There are 143 flue-tubes, 2 inches in diameter, placed in a 3 feet 10 inch barrel. This is a very proper limitation of the number of the tubes, though there is only ½ inch clearance between the tubes. It would have been better for the generation and circulation of the steam if the tubes had been only 1¾ inches in diameter, the same number being retained. Engines of this class are at work on several Welsh railways, and also at large ironworks and collieries.

V.—Double bogie tank-locomotive (Fairlie's system), designed and constructed by Mr. A. A. M'Donnell, for the Great Southern and Western Railway. Plate XLVII.

Area of Grate, ...	14·6 square feet.
Heating Surface (inside), . . .	738 ,,
Diameter of Cylinders, . . .	15 inches.
Stroke of Pistons, ...	20 ,,
Diameter of Driving-wheels, . .	5 feet 7½ inches.
Length of Wheel-base, extreme, .	25 ,, 7 ,,
Do. to centres of Bogies,	14 ,, 7 ,,
Do. of Leading Bogie,	6 ,, 0 ,,
Do. of Trailing Bogie,	5 ,, 0 ,,
Weight in working order, . . .	35 tons 17 cwts.
Do. on Leading Bogie, . .	22 ,, 0 ,,
Do. on Trailing Bogie, . .	13 ,, 17 ,,
Capacity of Water-tank, . . .	800 gallons.
Do. of Coal-boxes, . . .	30 cwts.

This engine is designed on the principle of the double bogie, which has been proposed and advocated from time to time, but only in recent years worked out, and become known as the system of Mr. Robert F. Fairlie. At the International Exhibition of 1862, the principle of applying the steam-power to a pair of bogies under one boiler, was

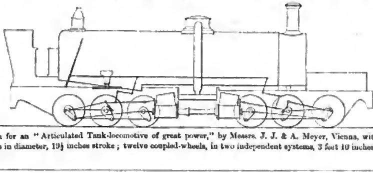


Fig. 224.—Design for an "Articulated Tank-locomotive of great power," by Messrs. J. J. & A. Meyer, Vienna, with four cylinders 17¼ inches in diameter, 19⅝ inches stroke; twelve coupled-wheels, in two independent systems, 3 feet 10 inches in diameter.

proposed and fully illustrated by M. Meyer of Vienna, as described at page 91 (*ante*).* (See also Fig. 224.)

An engine constructed on Mr. Fairlie's system, called the "Little Wonder," has been at work for some time on the Festiniog Railway, a line in Wales constructed to a gauge of 1 foot 11½ inches, or what is known in round numbers as the 2-feet gauge. The general arrangement of this engine is indicated by the annexed woodcut, Fig. 225. The boiler is double, having two fireboxes united back to back, with two distinct barrels and sets of flue-tubes, and consequently a chimney at each end. A bogie is placed under each barrel, and each bogie has two pairs of wheels, coupled together, worked independently by a pair of steam-cylinders to each bogie. Thus a total wheel-base of 19 feet 1 inch in length is covered by the bogies; each bogie has a 5 feet wheel-base, and the distance between the centres of the bogies is 14 feet 1 inch. The four cylinders are $8\frac{5}{16}$ inches in diameter, and have a stroke of 13 inches; the wheels are 2 feet 4 inches in diameter. The combined grate-area is 11 square feet, and the heating surface 730 square feet.

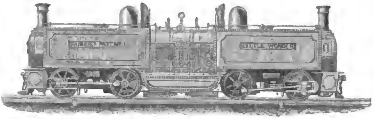


Fig. 225.—The "Little Wonder" Double Steam-bogie Engine, on Fairlie's system.

From the results of a series of well-attested trials of the "Little Wonder," made in the month of February, 1870, on the Festiniog Railway, the following may be selected for example. The "Little Wonder" left Portmadoc with a train of seventy-two waggons, 648 feet in length, as follows:—

	Tons	cwts.	qrs.
Loaded Waggons (slate trucks), weighing .	138	17	2
Empty Waggons, weighing	43	13	0
Passengers, weighing	4	0	0
Train Weight, ...	186	10	2
Engine Weight, ...	19	10	0
Gross Weight of Engine and Train, . .	206	0	2

The engine started with a steam-pressure of 170 lbs. per square inch, and drew the train up a gradient of 1 in 85, at a speed of five miles per hour. The train, when it was started, stood partly on a curve of 4½ chains radius, and partly on a reverse curve of 8 chains radius. The weather was fine, with a strong, cold head-wind. The rails weighed 30 lbs. per yard, and were not fished.

A comparative trial was made with the "Welsh Pony," an ordinary engine weighing 10 tons, with a pair of cylinders 8½ inches in diameter, 12 inches stroke, and 2-feet wheels. She was tried with a train of thirty waggons, but failed to start them on a gradient of 1 in 85. She was then tried with a train of twenty-six waggons, as follows:—

	Tons	cwts.	qrs.
Twenty-six Waggons, weighing . . .	62	6	0
Passengers, weighing	1	10	0
Train Weight, ...	63	16	0
Engine Weight, ...	10	0	0
Gross Weight of Engine and Train, . .	73	16	0

She started with the train on a gradient of 1 in 85, and partly on a curve of 4½ chains radius; and with an average pressure in the boiler of 150 lbs. per square inch, drew the train up the incline, at a speed of five miles per hour, for a quarter of a mile, when the train was purposely stopped.

At another trial with twenty-five waggons, making up a gross weight, including engine, of 72 tons 17 cwts. 2 qrs., the "Welsh Pony" drew the train up the incline, 1 in 85, having commenced with 140 lbs. steam in the boiler, and been stopped when the pressure fell to 130 lbs.; for what reason the engine was then stopped, there is no explanation given, though it would have been most useful, and it was in fact necessary, to ascertain the exact maximum performance of the "Welsh Pony" with a given pressure in the boiler, for the purpose of completing a proper comparison with the performance of the "Little Wonder."

Working out these performances in the usual way, and taking 76½ per cent. of the boiler-pressure as the effective mean pressure in the cylinders—which, for such low speeds as those at which the trials were conducted, is certainly not too much—the following comparative results are arrived at:—

	Total Resistance. Lbs. per Ton, gross.	Gravity. Lbs. per Ton.	Frictional Resistance. Lbs. per Ton.
"Little Wonder,"	40	26·3	13·7
"Welsh Pony," with 150 lbs. steam, .	51·4	26·3	25·1
Do. 130 lbs. steam, .	44 5	26 3	18·2

These comparative results of frictional resistance, in the last column, show a reduction from 25·1 lbs. and 18·2 lbs. per ton with the "Welsh Pony," to 13·7 lbs. per ton with the "Little Wonder." That is to say, in the one case, a reduction equal to 45 per cent.; and in the other, a reduction of 25 per cent. These results are incomprehensible. The reduction of resistance, whatever it may have been, could only have been effected in the locomotive itself;

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* See also the writer's book on *The Exhibited Machinery of 1862*, page 24, and *Proceedings of the Institution of Mechanical Engineers*, for 1863, in which he discusses at some length the merits and peculiarities of the locomotive engines at the International Exhibition of 1862. With respect to the problem of maximum-power engines, the writer, in 1863, pointed to the system of double steam-bogies, as then advocated by M. Meyer, as the proper method of working out the problem to the fullest extent. (See Fig. 224.) He says,—"This system of Messrs. Meyer, which they have so ably designed, affords unquestionably the most satisfactory solution of the problem of maximum-power goods engines. With independent steam-driven swivelling-frames, they get rid of a mass of coupling-rods and parallel motions, by which the resistance of the engine is materially reduced in its working parts, and in passing along curves leaving a balance of power for useful work." "In short, they employ two steam-bogies, which carry the boiler and its accessories on their backs, like a load of timber on a couple of swivelling timber trucks." (*Exhibited Machinery*, pages 21, 25.)

whereas the total frictional resistance of the "Welsh Pony," as an ordinary four-wheeled engine, could certainly not have amounted to 20 lbs. per ton of its proper weight, being equivalent to 2·75 lbs. per ton of the gross weight of the trial train. Now, suppose that the "Welsh Pony" had, by the application of Mr. Fairlie's system, been rendered even absolutely frictionless, the resistance could only have been reduced, in the first-named trial, to 25·1 — 2·75, equal to 22·35 lbs., or 11 per cent. less; in the second-named trial, to 18·2 — 2·75, equal to 15·45 lbs., or 15 per cent. less. But, dropping such an extreme supposition, and allowing that in practice a reduction of half the internal friction could be effected, by the application of the double steam-bogie system to the "Welsh Pony," then the resistance would have been reduced, in the first-named trial, to 25·1 — 1·38, equal to 23·72 lbs., or 5½ per cent. less; and in the second-named trial, to 18·2 — 1·38, equal to 16·82 lbs., or 7½ per cent. less. Thus, it appears that the resistance with the "Little Wonder" could not possibly, on any probable supposition, have been so low as 13·7 lbs. per ton gross, assuming, of course, that the two trains were in equally good order.

It may be added, that the "Progress," another engine on Mr. Fairlie's principle, was tried, in February, 1870, on the Mid Wales Railway. There are four cylinders, 15 inches diameter, with 22 inches of stroke, and 4 feet 6-inch wheels. The engine weighed 54 tons, and moved a train, consisting of forty loaded waggons and two vans, weighing 476 tons, up an inclined piece of road, about two miles long, averaging 1 in 115 of inclination. The engine and train ascended the incline at an estimated speed of about eight miles per hour, with 130 lbs. pressure in the boiler. The total resistance is estimated at 34½ lbs. per ton; from which, deducting the resistance due to the incline, 19½ lbs. per ton, there remains 15 lbs. per ton resistance of friction, which is greater than was calculated for the "Little Wonder." In the absence of an exact comparative trial with ordinary locomotives, it is difficult to draw any conclusion upon the comparative efficiency of the "Progress." A great deal can be done on an ordinary railway, with ordinary engines, with a traction force of 15 lbs. per ton.

Turning to the engine of Mr. M'Donnell, it differs from the "Little Wonder," and other engines on Mr. Fairlie's system, in having only one steam-bogie, the other bogie being without cylinders. The two bogies are connected by a carrying-frame which supports the boiler, as in all Mr. Fairlie's later engines. The second bogie is arranged on Mr. William Adams's plan, already described, page 283. The pivot of the leading bogie is formed with a flat face to rest upon the socket, with lateral india-rubber check-springs to regulate the traverse, as in Mr. Adams's bogie. Any tendency to pitch, on the part of the leading bogie, is checked by india-rubber springs fixed in front of the firebox. The bearing-springs of the leading bogie are connected by compensating springs; and for the second bogie, a single spring is applied on each side to span the two axles.

The engine is found to run with great steadiness, and it passes round curves of 300 feet radius with perfect ease. It can pass round a curve of 200 feet; but the driving-wheels then almost touch the carrying-frame.

Since the Fairlie engines just noticed were constructed, two have been built for Sweden; others are in progress (1870) for Russia; and one, the "Tarapaca," for the Iquiqui Railway, Peru, a line of 4 feet 8½ inches gauge. The "Tarapaca," of which the following particulars are from *Engineering*, is the largest engine yet constructed on the Fairlie system:—It is carried on twelve wheels, 3 feet 6 inches in diameter, arranged in groups of three pairs each, the wheels of each group being coupled. Each of the bogies has a pair of cylinders 15 inches in diameter, with 20 inches stroke. An effective mean pressure of 100 lbs. per square inch is expected to be maintained on the pistons, from which the tractive force, reduced to the line of rails, is estimated at 21,430 lbs., or about 9½ tons. The weight in working order is 60 tons, all available for adhesion, and is 6¼ times the gross traction force. The boiler is of the double form employed by Mr. Fairlie; the total length is 31 feet 6 inches. The two barrels are each 10 feet 6 inches long. The two fireboxes are separated only by a water partition. The heating surface in the fireboxes is 125 square feet; in the tubes, 1,500 square feet; making a total of 1,625 square feet. The fire-grate area is 21 square feet. The water-tanks hold 2,200 gallons, and the bunkers 2 tons of coal.

CHAPTER XXVIII.

EXAMPLES OF TENDERS.

1. Six-wheeled tender, by Mr. J. J. Cudworth, for coal-burning passenger engine, South-Eastern Railway. Plate XXVIII.

2. Four-wheeled tender, by Mr. William Cowan; constructed by Messrs. Neilson & Co., Glasgow, for the Great North of Scotland Railway. Plates XLI., XLII.

3. Steam tender, with six coupled-wheels, by Mr. Archibald Sturrock, for the Great Northern Railway. Plate XXV.

The first and second above-named tenders demand little explanation of their illustrative plates. They are examples respectively of six-wheeled and four-wheeled tenders, to carry respectively 1,600 and 1,070 gallons of water. The weight of the former, with 35 cwts. of coke, and filled with water, is 20 tons 10 cwts., or about 7 tons per axle. The weight of the latter is 16½ tons, or 8¼ tons per axle, with a full supply of coal and water.

In the six-wheeled tender the frame is of wrought iron, and consists of one longitudinal plate on each side connected together by seven transverse plates. The brake-gear is fixed altogether to the frame, and has no attachments of any kind to the tank,—which is good in principle. The water is held in a rectangular tank, with a straining well at the bottom, from which the water is taken to the pump. A branch from the exhaust-pipe of the engine is led into the tank at one side, so as, by the condensation of the steam, partly by the surface exposed, and partly by direct contact with the water, to raise the temperature of the feed-water. The connection with the engine is made by means of a transverse spring acting on two wrought-iron buffers, at 3-feet centres, moving in guides fixed to the tender, and bearing on the end frame-plate of the engine. The connecting draw-bar is 2 feet long, reaching 14 inches into the engine-frame. The pull on the train at the other end of the tender is similarly made through a short draw-rod, on a volute spring, the bearing of the spring being just 1 foot within the hind end of the frame. The wheel-base of the tender is 12 feet long.

In the four-wheel tender, on the contrary, the wheel-base is only 8 feet 6 inches, and, in the important matter of the arrangements for traction, there is more freedom for tractive action than in the six-wheel tender. The draw-bar is about 4 feet long, or twice the length of that of the other tender, and it penetrates the tender-frame 20 inches beyond the buffer heads, and 27 inches into the engine-frame. The long link of connection thus afforded is of material importance in mitigating the lateral friction of the engine and tender wheels in passing along curves, arising from the obliquity of the pull of the engine,—as compared with the shorter link in Mr. Cudworth's engine. Accordingly, whilst Mr. Cudworth places his engine draw-pin at a distance of 3 feet 10 inches from his hind driving-axle, Mr. Cowan has carried his draw-pin to within 17 inches of the hind driving-axle. With proper allowance for angular play, it is manifest that the connection of the latter engine with its tender has the advantage of greater freedom for tractive effort. Moreover, Mr. Cowan has, in the four-wheel tender, applied a draw-rod on a volute spring at the hind end, as in the six-wheel tender; but he has, also, brought forward the bearing of the spring to within 15 inches of the centre of the tender-frame,—employing a draw-rod penetrating 6 feet into the frame. The effect of those differences in the application of the traction-gear is, that whilst in the six-wheel tender the two points through which the traction force is exerted and transmitted are 17½ feet apart, in the four-wheel tender they are only 8 feet 9 inches apart, or exactly half the distance. Though this difference is in part accounted for by the greater length and capacity of the six-wheel tender, yet it is for the most part the effect of design,—the wise substitution of polygonal lines of tractive force for ordinary angular connection by the extremities.

The frame of the four-wheel tender consists of a pair of longitudinal plates on each side, connected together by six transverse plates, with a plated wooden buffer-beam in addition. The side chains pull upon volute springs. The water-tank is of the ordinary horse-shoe form, which, it must be said, is the most comfortable and convenient form of tank. The brake-rod is conducted at an angle down the inside of the tank, to take the screw-nut directly; and though it is better thus, in the four-wheel tender, for the sake of direct action, yet the system of Mr. Cudworth, of a vertical screw-rod with an equally direct connection, and with all the bearings on the frame, is preferable.

The steam tender designed by Mr. Sturrock, Plate XXV., represents a class of practice which is struggling into existence, and which was pretty widely represented at the French Exposition of 1867; and probably the principle of auxiliary steam power thus represented is more highly appreciated on the Continent than it is in England. The steam tender of Mr. Sturrock—though it has been abandoned on the Great Northern Railway, for which line it was originally devised, to increase the tractive power of goods trains on inclines—deserves to be remembered as a contribution to the practice of auxiliary power. The tender, thus converted, becomes in fact a locomotive engine with a water-tank instead of a steam-boiler. It has six wheels, and is fitted with a pair of inside-cylinders, 12 inches in diameter, with 17-inch stroke, connected to the middle axle, which is cranked; and the three axles are coupled by means of outside rods. The wheels are 4 feet in diameter. The steam is conducted to the tender through a long pipe, from the boiler of the engine, about 23 feet in length, with five bends, which yields elastically to the various inflections to which it is subjected, without the intervention of compensation joints. The exhaust steam from the cylinders is delivered into a tubular condenser, surrounded by the water in the tank, consisting of fifteen tubes, 2 inches in outside diameter, about 12 feet 8 inches in length, fixed into a reception-box at each end. The first box receives the exhaust steam, and delivers it through the tubes; the second is fitted with a waste-pipe to carry off the uncondensed steam. The condensed steam, or water formed in the condenser, appears to have had no other exit than by the waste-pipe, through which it must have been projected by the uncondensed steam; and in order to prevent the escape of such projected water, a baffle-plate and an inverted cone were inserted at the exit of the pipe. Fifty steam tenders of this design were at one time at work on the Great Northern line. The weight of the tender, empty, was 18 tons; full, 27¾ tons.

CHAPTER XXIX.

RESISTANCE OF LOCOMOTIVES AND TRAINS.

The best series of recorded observations on the resistance of railway trains was made by Sir Daniel Gooch, in 1847, upon the Bristol and Exeter Railway, 7 feet gauge. They were conducted in a spirit of the strictest impartiality; and as the experimental applications were elaborated with the utmost care and consideration, the record of his observations may be accepted with confidence.*

"I selected," says he, "a mile of line, perfectly straight and level, and nearly on the surface of the ground, a plan and section of which accompanies these experiments, and upon which the height of hedges and trees are marked as affecting the action of the wind upon the train. To judge of the force of the wind during each experiment, as distinct from that registered by the dynamometer, I noticed the effect it had upon some willows growing on the side of the line, as explained in the sheet containing the diagram of the experiments.

"It was my intention, on commencing the experiments, to make four with 100 tons each, at speeds of twenty, forty, and sixty miles per hour, and the same number with 50 tons. The experimental train consisted of first and second class carriages, on six wheels, 4 feet in diameter, taken indiscriminately from the working stock of the Company, and loaded with iron, to represent a fair load of passengers. This gives an average gross weight for each coach of 10 tons. The weather during the four days was not very favourable, there being a great deal of wind and rain. All these particulars were, however, very carefully noted, and will have to be taken into consideration in determining the correct resistance of each experiment. It was also found impossible to obtain exactly the speed required, or yet to get the speed perfectly uniform over the trial mile, and for a short distance before coming to it,—allowance will, therefore, also have to be made for any acceleration or diminution of velocity during the experiments, which I, however, fear is a very difficult amount to estimate. In the tables accompanying this paper [and in Report], the various experiments are arranged in series, according to the weight of the train and the speed." A series of experiments were also made on Brimscombe incline, with various trains and speeds.

The primary results of these valuable experiments were reduced, condensed, and arranged by the present writer in 1854; and he at the same time constructed curves, and deduced formulas, and tables, based on these results, for the resistance of engines, tenders, and trains.† To reduce and utilize the observations made with accelerated or retarded speeds, alluded to by the experimentalist, the writer deduced and employed the formula—

$$x = \frac{V^2 - V'^2}{70\ D}.$$

In which V and V' are the initial and final velocities, in miles per hour, on the measured distance; D is the measured distance in miles; and x the accelerating or retarding force, according as the final speed was greater or less than the initial speed, in lbs. per ton.

The resistances thus reduced for uniform speeds were thrown into curves of resistance, of which those for the 100-tons trains are illustrated by Figs. 226, 227, and 228, for which the base lines represent the speed, up to sixty miles per hour, and the vertical ordinates, the resistance in lbs. per ton. The cross marks indicate the result of each observation, showing the total amount of the resistance per ton, as moving at the given uniform speed. In all these diagrams, it will be found, the resistances are divisible into two parts: the constant and the variable

Resistances with 100-tons train, in lbs. per ton.

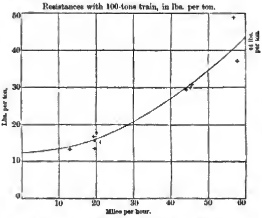

Fig. 226.—Proper Resistance of the Engine and Tender.

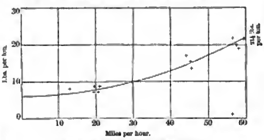

Fig. 227.—Proper Resistance of the Train.

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* "On the Resistance to Railway Trains at Different Velocities:" a paper read by Sir Daniel, then Mr. Gooch, before the Institution of Civil Engineers, 18th April, 1848.

† See *Railway Machinery*, page 292.

resistance, of which the latter varies as the square of the speed, starting from nothing when the train is just in motion. The constant resistance is due to the internal friction of wheels, axles, and machinery; the variable resistance is due to the atmosphere, to lateral oscillation and concussion, to vertical oscillations and concussions between the wheels and the rails, due to imperfections of the permanent way, to the coning of the wheels, and their consequent sledging on the rails at high speeds, and to other cognizable causes. All these naturally vary as the square of the speed;—none of them as the speed simply, though this has been assumed by Mr. Scott Russell for certain resistances. The curves of resistance, however, embracing the resultant resistance from all sources, prove that, as a whole, the variable resistances increase as the square of the speed.

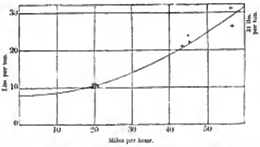


Fig. 228.—Total Resistance of the Engine, Tender, and Train.

For facility of reference, the following statement is given, showing the resistances at extreme speeds:—First, the constant resistance when the engine or train is just put in motion; and second, the resistance at sixty miles per hour. The former, the constant resistance, must be overcome before any motion at all can be produced; and the latter comprises the constant and the variable resistances together

	At 60 miles per hour.
1st.—ENGINE and TENDER alone:—	
With 100-tons train, 12¾ lbs. per ton, constant,	44 lbs. per ton.
With 50-tons train, 14¾ lbs. per ton, constant,	40 ,, ,,
2nd.—TRAIN only:—	
100-tons train, 6¼ lbs. per ton, constant,	21½ lbs. per ton.
50-tons train, 8 lbs. per ton, constant, . .	23 ,, ,,
3rd.—ENGINE, TENDER, and TRAIN, total:—	
With 100-tons train, 8 lbs. per ton, constant,	31 lbs. per ton.
With 50-tons train, 10 lbs. per ton, constant,	32 ,, ,,

It is observable that in each case the constant resistance with the 50-tons train is about 2 lbs. per ton more than with the 100-tons train. This excess is due to the state of the rails, which were, with the 100-tons train, in all cases dry and clean; but with the 50-tons train, were dirty and wet, in seven cases out of twelve. But, though the constant resistance was thus raised, it appears that the state of the rails did not affect the variable resistance; since the variable resistance per ton, at sixty miles per hour, is virtually the same for both trains; being

For the 100-tons train, 21½ — 6¼ = 15¼ lbs. per ton.
For the 50-tons train, 23 — 8 = 15 ,, ,,

There is no doubt that had the rails been in equally good order, the constant resistances also would have been the same.

The following are the formulas and rules for the resistance of engines and trains, &c., constructed by the writer from Sir Daniel Gooch's data. They are based on the experimental results with the 100-tons train; and the conditions are—

1st. The permanent way in good order.
2nd. The engine, tender, and train, in good order.
3rd. A straight line of rails.
4th. Fair weather, and dry and clean rails.
5th. An average side-wind, of average strength, varying (in the experiments) from *slight* to VERY *strong*.

Let V = the given speed.
R = the total resistance of the engine, tender, and train, per ton gross.
R' = the resistance of the train alone, per ton.
R'' = the resistance of the engine and tender alone, including machinery friction.
W = the weight of the engine and tender.
w = the weight of the train.

RULE I.—To find the total resistance of the engine, tender, and train, at a given speed. Square the speed in miles per hour, divide it by 171, and add 8 to the quotient. The sum is the total resistance reduced to the rails in lbs. per ton gross weight. Or, by formula,

$$R = 8 + \frac{v^2}{171}. \quad . \quad . \quad . \quad (1.)$$

RULE II.—To find the resistance of the train alone, at a given speed. Square the speed in miles per hour, divide it by 240, and add 6 to the quotient. The sum is the resistance reduced to the rails, in lbs. per ton weight of train. Or, by formula,

$$R' = 6 + \frac{v^2}{240}. \quad . \quad . \quad . \quad (2.)$$

RULE III.—To find the whole resistance of the engine and tender alone, with a given train, at a given speed. 1st. Find the resistance of the engine and tender, as carriages, by RULE II. 2nd. Find the machinery friction, thus:—Square the speed in miles per hour, and divide it by 600, and add 2 to the quotient. Multiply the sum thus found by the gross weight of engine, tender, and train, and divide by the weight of the engine and tender alone. The quotient is the machinery friction. 3rd. Add together the two results thus obtained. The sum is the resistance of the engine and tender reduced to the rails in lbs. per ton weight. Or, by formula,

$$R'' = 6 + \frac{v^2}{240} + \left((2 + \frac{v^2}{600}) \times (\frac{W + w}{W})\right). \quad (3)$$

The foregoing rules are, it may easily be understood, applicable under a set of conditions more favourable than the actual average conditions of work on railways. And in order to arrive at a formula of resistance which could safely be employed, in ordinary practice, so as to meet the unfavourable conditions in combination, of frequent quick curves, under one mile radius, and strong side and head winds, the writer estimated, from his own observations, that the resistance, as calculated by means of the foregoing rules, should be increased 50 per cent., or one-half more. A formula which would express such increased total resistance of the engine, tender, and train, similar to formula (1) above, would comprise a constant of 12 lbs. per ton, and for denominator of the fraction, 114; and it would stand thus:—

$$R = 12 + \frac{v^2}{114}, \quad . \quad . \quad . \quad (4.)$$

The annexed table contains the total resistance of engine, tender, and train, calculated by means of this formula (4).

RESISTANCES PER TON OF ENGINE, TENDER, AND TRAIN, AT UNIFORM SPEEDS, AND ON VARIOUS GRADIENTS, MEASURED BY TRACTIVE FORCE AT THE RAILS.

Ascending Gradients.	Conditions:—A Good Road; Frequent Curves under One Mile Radius; Strong Side and Head Winds; Engine, Tender, and Train in Good Working Order.										
	Speeds in Miles per Hour.										
	10	15	20	25	30	35	40	45	50	55	60
	Total Resistance, in Lbs. per Ton, due to the above Speeds and the annexed Ascending Gradients; Constant, 12 Lbs.										
	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
Level	13	14	15·5	17·5	20	22·5	26	30	34	38·5	43·5
1 in 20	125	126	128	130	132	135	138	142	146	151	156
1 „ 25	103	104	105	107	110	112	116	120	124	128	133
1 „ 30	88	89	90	92	95	97	101	105	109	113	118
1 „ 40	69	70	71	73	76	78	82	86	90	94	99
1 „ 50	58	59	60	62	65	67	71	75	79	83	88
1 „ 60	50	51	53	55	57	60	63	67	71	76	81
1 „ 70	45	46	47	49	52	55	58	62	66	70	75
1 „ 80	41	42	43	45	48	50	54	58	62	66	71
1 „ 90	38	39	40	42	45	47	51	55	59	63	68
1 „ 100	35	36	38	40	42	45	48	52	56	61	66
1 „ 110	33	34	36	38	40	43	46	50	54	59	64
1 „ 120	32	33	34	36	39	41	45	49	53	57	62
1 „ 130	30	31	33	35	37	40	43	47	51	56	61
1 „ 140	29	30	31	33	36	38	42	46	50	54	59
1 „ 150	28	29	30	32	35	37	41	45	49	53	58
1 „ 160	27	28	29	31	34	36	40	44	48	52	57
1 „ 180	26	27	28	30	33	35	39	43	47	51	56
1 „ 200	24	25	27	29	31	34	37	41	45	50	55
1 „ 250	22	23	25	27	29	32	35	39	43	48	53
1 „ 300	20	21	23	25	27	30	33	37	41	46	51
1 „ 350	19	20	22	24	26	29	32	36	40	45	50
1 „ 400	18	19	21	23	25	28	31	35	39	44	49
1 „ 500	17	18	20	22	24	27	30	34	38	43	48
1 „ 600	17	18	19	21	24	26	30	34	38	42	47
1 „ 800	16	17	18	20	23	25	29	33	37	41	46
1 „ 1000	15	16	18	20	22	25	28	32	36	41	46

It is worthy of special observation that the resistance of trains, adopting as a standard for comparison the current practice of trains in England, may be considerably modified by the way in which the train is placed on its wheels—for example, placing the train on bogie-frames and wheels—and by the employment of oil for the lubrication of the train instead of grease. The best performances of grease-lubricated carriages and waggons have never reached so low a resistance as those of the Great Western experimental trains, showing an initial resistance to traction of only 6 lbs. per ton. Observations made by the writer in 1850, on the resistance of waggons, show an extraordinary diversity of resistance,—varying from 7½ lbs. per ton of the train alone, and 13 lbs. per ton, including the engine, to 29 lbs. per ton of engine, tender, and train, at speeds under thirty miles per hour. This extreme diversity is no doubt due to the state of repair of the waggons, and of the way, the state of the lubrication, and the curves on the line.

Again, in the working of two coal-trains of waggons having inside bearings, the following diversity was shown in the gross resistance of engine, tender, and train:—

1. At 11 miles per hour, 27 lbs. per ton.
2. At 13 miles per hour, 18 „ „

The second train had only two-thirds of the resistance of the other; and the advantage was no doubt due chiefly to the lateral play of the journals in the bearings, in the second case, by which the wheels were free to adjust themselves without effort to the rails, swaying towards either side, as they might happen to be biassed by the conical tyres and unlevel rails. In the first case, the want of such play prevented the wheels from so adjusting themselves laterally without, at the same time, carrying with them the superincumbent loads. For the moderate speeds at which the trains were run, lateral play at the bearings is undoubtedly beneficial.

But the great practical improvement of the day, at least so far as English rolling stock is concerned, is to be the substitution of radial axles and axle-boxes, as those of Mr. W. B. Adams, or of bogie-frames and wheels, for the rigid wheel-bases of carriage and waggon stock:—bogie-frames, that is to say, in so far that individual waggons should be manipulated in the train as bogies, if not even to the extent of placing waggons of a larger class on a pair of bogies;—and rigid wheel-bases, that is to say, in so far that in common practice the axles of a carriage or a waggon are put in and maintained in positions parallel to each other, without the option of swivelling to curves. The practice of the United States, in the universal adoption of bogies under the engines, tenders, carriages, and waggons, is well worthy of analysis; and whilst it is maintained, with a considerable degree of truth, that the bogie is essential on the imperfect roads of America, to enable the wheels to follow the rail, and thus to obviate the chances of derailment, it must also be affirmed that, by means of the bogie, the tractive resistance of engines and trains is notably less than that of a parallel-axled stock.

In connection with the bogie, or the radial axle, the general use of oil for lubrication of axles further reduces the resistance to traction, as compared with the use of grease; and there are therefore, in the immediate future, two things to be done,—the general adoption of the radiating system in rolling stock, and the general substitution of oil for grease in lubrication.

In the application of the bogie principle to waggons, it may not, in many cases, be advisable to place two distinct bogies, with eight wheels, under one waggon; but at least an equivalent adaptation may be effected, by applying the tractive force to a central point in the waggon: it may even be a solid central pivot, from which point the waggon is to be drawn, and by which point it is to be connected to the following waggon. Were such a system of traction generally employed, in place of the common method of seizing an overhanging fixed drawhook at each end of the waggon, the necessary tractive force would be very materially reduced, on straight lines as well as on curves; for, whilst the waggon would continue to be the medium for the transmission of tractive force, it would be left at full liberty to play over the rails, and suit itself to irregularities—thus displacing a quantity of sledging action, arising from the unavoidable binding of rolling stock on the rails, when coupled in the usual manner, by substituting a movement more nearly akin to perfect rolling. But in whatever form the radial principle, or the principle of the bogie, is to be worked out in English rolling stock, the fact is certain, that the resistance to traction will be reduced, and the durability of the stock increased. M. Zerah Colburn describes very specifically a course of experiments which he conducted with a goods engine and train, wheeled on the bogie system, on the New York and Erie Railroad, over a total distance of 900 miles, on various gradients. "The engine had four coupled-wheels and a bogie, the total weight in working order being 29½ tons, of which 17¾ tons rested on the coupled-wheels available for adhesion. The coupled-wheels were 5 feet in diameter, the outside cylinders were 17 inches in diameter, and the stroke 24 inches. The safety-valves were set to blow off at 130 lbs. per square inch; and the steam, as observed by a Bourdon gauge, was seldom allowed to exceed that limit." "The tender, loaded, weighed 18½ tons. The train consisted of eight-wheel waggons, fully loaded with deals,—the average weight of each waggon, empty, being 5 tons 8 cwts. 3 qrs., and of each waggon with its load, 15 tons 5 cwts. 3 qrs. nearly. The waggons had cast-iron chilled wheels, 2 feet 6 inches in diameter, with inside journals 3¾ inches in diameter, and 8 inches long. All the waggons had been put in complete order, and the journals, fitted with oil-tight boxes, were kept well oiled. The line had a gauge of 6 feet. The weather was most favourable, clear and dry, with the exception of a single day of heavy rain.

"Upon about 100 miles of the line, forming a portion of the Susquehanna division, a train of 100 waggons, weighing, with engine and tender, 1,572 tons, was taken. The train was a few feet more than half a mile in length."

"At one point it was stopped, where the line commenced an ascent of 24 feet in four miles, averaging 1 in 880 up for the whole distance. There were also long and easy curves upon this portion. The train was taken up, and purposely stopped, on the second mile, to be sure of starting again with no aid from momentum. The average speed up was five miles an hour, and neither was the pressure of steam increased, nor sand used, except in starting from the stops purposely made. The engine, even were its full boiler pressure of 130 lbs. maintained as effective pressure upon the pistons throughout the whole length of their stroke, could not have exerted a tractive force greater than

$$\frac{17 \times 130 \text{ lbs.} \times 2 \text{ ft.}}{5 \text{ ft.}} = 15{,}028 \text{ lbs.};$$

nor is it at all probable that the effective cylinder-pressure could have approached this limit by from 10 lbs. to 15 lbs. per square inch. Supposing, however, for the sake of a *reductio ad absurdum*, that the full boiler pressure had been maintained upon the pistons for the whole length of their strokes, the adhesion of the coupled driving-wheels, not deducting the internal resistances of the engine, would have been $\frac{15028}{39760} = \frac{3}{8}$ths of the weight upon them. In any case there was a resistance of 4,011 lbs. due to gravity; and if even 120 lbs. mean effective cylinder pressure be assumed, corresponding to a total tractive force of 13,872 lbs., the quotient, representing the rolling and other resistances, exclusive of gravity, would be but 6·27 lbs. per ton of the entire train,—a resistance including all the internal resistances of the engine, the resistance of the curves, easy as they were, and the loss in accelerating and retarding the train in starting and stopping. This estimate of resistance would correspond, at the observed speed of five miles an hour (upwards of three-quarters of an hour having been consumed on the four miles), to 185 indicated horse-power, which, with the driving-wheels making but twenty-eight revolutions per minute, would be the utmost that an engine with but 1,038 square feet of heating surface could be expected to exert. This was the highest result observed during the three weeks' trial; but one or two others are worthy of mention. On the Delaware division of the same line, the train of 1,572 tons' weight was run over five consecutive miles of absolutely level line, at a mean rate of 9·23 miles an hour; and, during the same day, over five other consecutive miles of level, at a mean rate of 9·7 miles per hour. On both levels there were 14½ chain curves of good length; and the speed—from nine to twelve miles an hour—at which the train entered the respective levels, was not quite regularly maintained throughout the half hour expended in running over them. But if even 7 lbs. per ton of the total weight be taken as the resistance of these speeds, the tractive force will be 11,004 lbs., which is more than one-fourth the adhesion-weight of 40,050 lbs. On the next day, the same engine drew thirty waggons, weighing 466½ tons, or, including engine and tender, 514 tons nearly, up a gradient of 1 in 117½, three miles long, at a mean speed of 10½ miles an hour. The resistance due to gravity was 9,814 lbs.; and, supposing the other resistances to traction to amount to no more than 7 lbs. per ton, the total resistance would be 13,412 lbs., corresponding to a mean effective cylinder pressure of 117 lbs. per square inch, and to a co-efficient of adhesion of almost exactly one-third."

There is nothing very improbable in these results of the performance of the engine, when it is considered that every waggon was placed on double bogies, though, it must be confessed, with Mr. Colburn, that the result is in some respects unique in the history of railway traction. He has estimated, from his experiments, that the gross resistance of the engine, tender, and train, could not have been more than from 6½ to 7 lbs. per ton, at a speed of from five to ten miles per hour; in comparison with which, the gross resistance of the Great Western engine and train, at ten miles per hour, was 8½ lbs. per ton on a straight line, with average side-winds.

In corroboration of the results of Mr. Colburn's experiments, Mr. E. A. Cowper has produced the results of experiments he made, to test, by means of a dynamometer, the tractive resistance of ordinary carriages of the London and South-Western Railway, fitted with Mr. Beattie's close oil axle-boxes; in some cases one carriage only, and in others two carriages, being towed. The force required to start the carriage amounted, in some cases, to 4 lbs. per ton, and in others the friction was only 2·1 lbs. per ton, to keep the carriage just in motion, when the rails were level. In another case, a force of 2 lbs. per ton of the gross load was sufficient to keep the carriage just in motion, on level rails. At a speed of fifteen miles per hour the friction was 2·4 lbs. per ton; and at twenty-five miles per hour it was 2·8 lbs. per ton. Similarly, the friction of waggons weighing from 6 to 7 tons, with white metal bearings and close oil-boxes, averaged 2·85 lbs. per ton.

The effect of the foregoing evidence is, that a constant resistance of 8 lbs. per ton, which has usually been adopted in calculations of the resistance of trains running in grease, is likely to be succeeded by a resistance at least a half less, and that in the formula for resistance of trains, the element of constant resistance will be correspondingly reduced by the substitution of oil for grease in lubrication. The effect of that substitution, however, will end there, as the variable resistance will not be affected by it; and it behoves engineers not to deceive themselves with a prospect of materially reducing the cost of engine-power for haulage, however it may reduce the cost for lubrication. The great source of economy lies in the substitution of the principle of the bogie for the sledge-like action of fixed parallel axles; for, by that means, not only the constant element of resistance, but likewise the variable resistance, will be materially diminished, and will bring with it a material reduction in the cost of maintenance, as well as of engine-power. All experience shows that the grinding of wheels fixed on parallel axles, especially the leading-wheels of engines in passing along quick curves, absorbs a good deal of power: that must be the case wherever there is a long wheel-base with parallel axles. The writer found that, with a passenger-train on the Caledonian Railway, going forty-five miles per hour, the resistance rose from 28 lbs. to 34 lbs. per ton gross, as measured by the indicator on the cylinder when the train passed from a portion of the line which was practically straight to a portion of the line having curves of less than one mile radius, but more than half a mile radius, averaging 2½ curves per mile. The increase, in this instance, of resistance, caused by the curves, was more than 20 per cent. of the resistance on the straight part of the line. This great extra resistance must be caused by, more than anything else, the obliquity of the axles, or their want of squareness with the rails; and it is evident that where, by the use of the bogie system, the axles are brought into the desired state of squareness with the rails, particularly the leading axle of the engine, the resistance on curves must be comparatively approximated to that on straight lines. Any one who has observed a heavy-laden waggon turning a street corner, grinding on the kerb-stone, must have been struck with the great strain in pushing the wheels sidewise; and it is precisely this lateral straining, or surging, as Mr. W. Bridges Adams describes it, involving extra resistance, that takes place every time an engine with parallel axles passes along a curve. End play in the axle-boxes, it is true, prevents binding on curves; but this can be better done by widening the gauge, and it does not mitigate the chief objection to parallel axles. The writer has observed that, for engines in passing along the quick curves of sidings, the resistance has been more than twice as much as upon a straight line. A good means of judging of the resistance of different forms of engines is to compare their performance on steep inclines (with which quick curves are usually associated), for which purpose the results of the performance of a number of engines on inclines are given in the annexed table, abstracted partly from published statements, and partly from the observations of the writer. The performances are reduced, for the sake of a direct comparison, to the equivalent performance on the steepest incline on the list—namely, 1 in 27, on the Mauritius Railway. It appears that the Semmering engines, having coupled parallel axles, take in gross an equivalent of about three and a half times their adhesion-weight. The Giovi engines, being two tank engines coupled together, take an equivalent little more than two and a half times their adhesion-weight. The Mauritius engines, having coupled parallel axles, take three times their adhesion-weight. The coupled engines on the Santander Railway, with a bogie in front, take an equivalent four times their adhesion-weight. The four-coupled-wheel tank engines (having two axles 8½ feet apart), on the Great North of Scotland Railway, take up

COMPARATIVE PERFORMANCES OF INCLINE ENGINES.

	Name or Locality.	Adhesion-weight.	Weight of Engine and Tender.	Weight of Train.	Gross Weight of Engine, Tender, and Train.	Ascending Gradient.
		Tons.	Tons.	Tons.	Tons.	
1	Semmering,	37·5	59	130	189	1 in 40
2	Do.,	46·5	66	175	241	1 „ 40
3	Do.,	37	56·4	100	156·4	1 „ 40
4	Great North of Scotland Railway—Bogie and Tender,	17·5	43	189	232	1 „ 75
5	Do. do. do.	„	„	144	187	1 „ 50
6	Do. do. Tank Engine,	25	25	209	234	1 „ 59
7	Do. do. do.	„	„	231	256	1 „ 59
8	Giovi (two Tank Engines coupled together),	55	55	100	155	1 „ 29
9	Mauritius Railway,	48	48	100	148	1 „ 27
10	Santander Railway,	36	45	200	245	1 „ 50

	Gross Weights or Loads Reduced for a Gradient of 1 in 27.					
1	135 tons Gross	=	3·4	3·6 times.	37·5 tons	Adhesion-weight.
2	173 „	=		3·72 „	46·4	„
3	112 „	=		3·02 „	37	„
4	103 „	=	6¼	6·0 „	17·5	„
5	112 „	=		6·4 „	17·5	„
6	123 „	=	5¼	5·0 „	25	„
7	135 „	=		5·4 „	25	„
8	145 „	=		2·6 „	55	„
9	148 „	=		3·0 „	48	„
10	147 „	=		4·0 „	36	„

five times their adhesion-weight; and the four-coupled-wheel goods engines, with a bogie in front, on the same line (of a class similar to those illustrated in Plates XLI., XLII.), take up six times their adhesion-weight. The amounts given are averages: they clearly show that the bogie engine is more efficient than the ordinary multi-coupled engine without bogies. The comparatively small performance of the Giovi engines is a consequence of the ordinary but defective mode of coupling two engines together—namely, a short central coupling with a great overhang from the wheel-base, which induces a great leverage to bind the engines hard against the inner rail on curves. The advantage of the bogie is very directly shown in comparing the two engines on the Great North of Scotland Railway. The four-coupled-wheel tank engines weigh 25 tons, all adhesion-weight; and they move an equivalent gross load, including their own weight, of 130 tons. The bogie engine, with only 17½ tons adhesion-weight, moves an equivalent gross load of 110 tons; that is, with about two-thirds adhesion-weight, they take six-sevenths of the gross load taken by the tank engines.

CHAPTER XXX.

SUPPLEMENTARY CHAPTER ON COAL-BURNING IN LOCOMOTIVES.

The performances of locomotives, having boilers specially designed for the combustion of coal, represented by those of Mr. M'Connell, Mr. Beattie, and Mr. Cudworth, have already been compared and tabulated in the notice of Mr. Cudworth's locomotive, at page 265. Some more specific notice than has yet been accorded to them appears to be wanted, of such coal-burning appliances as have been adapted to the locomotive boiler of the ordinary form; that is to say, of the boiler as it was constructed primarily for burning coke. Mr. Colburn, it is true, briefly notices (page 93) the baffle-plate, now (1870) commonly applied inside the doorway, for the purpose of deflecting air upon and amongst the fuel and the combustible gases in the firebox; and also the air-tubes originally applied by Hall on the Midland Railway. Mr. Kohn, too, has investigated the general principles of combustion, page 213. In this chapter it is intended, though at the risk of partial repetition, to present at one view the whole principle and practice of coal-burning in locomotives.

The physical conditions of the complete combustion of coal are,—*first*, that the introduction of air should be in sufficient quantity, and suitably distributed amongst the solid and gaseous portions of the fuel in the furnace, altogether or partially, through the grate, and partially above the fuel, directly to the gases; *secondly*, that the temperature should be maintained sufficiently high within the furnace, so that it should not be lowered by external causes during the combustion of the hydro-carbon gases, in order to effect the union of the element of carbon with its full proportion of oxygen; and *thirdly*, that the combustible gases should be thoroughly mixed with their supply of air.

With the limited firebox of a common locomotive boiler to deal with, enginemen have had recourse to various ways of treating the fire, in order to diminish the nuisance of smoke. They have relied chiefly on the instrumentality of the ash-pan, the dampers, and the fire-door, for carrying out a system of careful firing. They have endeavoured to prevent the formation of smoke, by controlling the admission of air through the grate, and adjusting it precisely to the requirements of the fuel by similarly manœuvring the fire-door for the admission of air above the fuel; by stoking with large pieces of coal and deep fires for heavy duty, and smaller coal with shallow fires for lighter duty; by firing more frequently, to lighten the duty; and at all times, by keeping the bars covered with fuel, to prevent excessive local draughts through the grate.

That much may be effected by these means, there is no doubt. It is well understood that the nuisance of smoke on entering or waiting at stations may be very much, or in some cases even altogether, subdued by thoroughly closing the ash-pan, and by opening wide the fire-door.

Low pitched doorways, at the level of the fuel, introduce fresh air to greater advantage than those which are set high, as the air through the low door mixes more quickly and more freely with the gases as they rise. It was accordingly found advantageous to adopt the practice of charging the fresh coal chiefly under the fire-door upon the hind part of the grate, and subsequently, when relieved of its gaseous element, pushing it forward towards the tubes, thus making way for a succeeding charge behind. Such have been the usual modes of treating coal, as fuel, in locomotive boilers of the normal type, in their normal condition. But the elaborations of practice in stoking ordinary boilers have been to a considerable extent superseded within the last few years by the methods recently introduced, of admitting air above the fuel, near its surface, and amongst the combustible gases. These methods may be classed as acting—*first*, by currents of air introduced through tubular or other openings in the sides of the firebox, uniformly distributed over the surface of the fuel; and *secondly*, by the deflection of a body of air introduced through the doorway, upon and over the surface of the fuel. A different principle of operation has also been introduced, by constructing large and spacious fireboxes, with large grates and long runs.

Of the three leading classes of coal-burning boilers, thus indicated, the first in chronological order is the adaptation of large extended fireboxes and combustion-chambers by Mr. M'Connell, in 1853; by Mr. Beattie, in 1855; and by Mr. Cudworth, in 1857. These have previously been sufficiently described and compared.

But it was necessary to adapt the existing engines, as they were, for burning coal without smoke by simple means, and independently of extensive structural alterations. This had been attempted by Messrs. Gray & Chanter in 1837, and again in 1839. They divided the firebox into two compartments—one for coal, the other for coke; they also admitted air in streams through tubes in the walls of the firebox; and they were the first who applied a steam-jet in the chimney, when the blast was off, to maintain a draught. Mr. Dewrance, in 1845, divided the firebox into two parts—the furnace and combustion-chamber. Messrs. Dubs & Douglas, in 1856, constructed a deflecting midfeather, proceeding from the back of the firebox towards the tube-plate, to throw down the smoke over the incandescent fuel; and Messrs. Evans & Dubs, in 1857, added to this a movable inclined grate, capable of being raised or lowered for the management of the fire. After various attempts, Mr. Yarrow, in 1857, projected an arch of brick from the tube-plate towards the fire-door, and admitted air through tubes in the tube-plate under the arch, and through the fire-door to mix with the smoke. He subsequently superseded the air-tubes by upright air-bars near the tube-plate. In the same year Mr. Jenkins, admitting air through numerous air-tubes in the tube-plate, applied a curved partition or baffler across the firebox to throw forward the air, distributing it through numerous small holes at the upper part. He regulated the supply of air by a damper, and more recently he added a row of air-tubes under the fire-door.

Towards the end of 1857, the writer, believing that a simple means of increasing the supply of air, and forcibly distributing and mixing it with the smoke in a common open firebox, independently of internal appliances, was a desideratum, devised his system of steam-inducted air-currents, in which, in its simplest form, air-currents are admitted just above the fuel, by tubes or otherwise, through the sides of the firebox, and are forcibly accelerated by means of jets of steam directed from the outside through the openings into and across the firebox; the steam nozzles operating in the air-tubes, as the blast does in the chimney. In December, 1857, the writer applied his system to one side of a small stationary-locomotive boiler at the railway foundry, New Cross. There were two air-tubes, and a jet of steam to each tube, from small nozzles on the outside, 1-16th of an inch in diameter, to increase the quantity and velocity of the air introduced. This simple apparatus was for several months in operation, and the dark smoke which was discharged when it was not in action was entirely prevented by its use. In January, 1858, a tank-engine on the North London Railway was fitted with four air-tubes, 1½ inch in diameter, through one side only, the other being inaccessible; with the assistance of jets of steam of 1-16th of an inch, smoke was completely prevented when the surface of the fuel was below the level of the air-tubes. In April, 1858, a passenger-engine on the Eastern Counties Railway was fitted with a row of air-tubes on each side,—four on one side, and three on the other,—with steam nozzles, one to each air-tube. With this application, the engine worked the main-line trains for eight or nine months, and the smoke was in general effectually prevented. In January, 1859, a passenger-engine on the South-Eastern Railway was fitted and put to work with two rows of air-tubes through the front and back walls of the firebox,—seven in each row, or fourteen in all, with induction-nozzles to every tube. On the Great North of Scotland Railway an engine was fitted, on the writer's system, with front and back air-tubes, under the direction of Mr. Cowan, the locomotive superintendent, and was started in March, 1859. The results were in all respects satisfactory; and the same system has been applied by the superintendent to the entire locomotive stock of that railway. (See Plates XLI., XLII.) In 1859, one of the old passenger-engines of the London and Brighton Railway was fitted with the author's apparatus, having a row of air-tubes and induction-jets in the front only. Accidental circumstances rendered impracticable the insertion of air-tubes in the back. Smoke was effectually prevented, except when heavy charges of coal were delivered at once.

On the Great Western Railway, coal-smoke has been burned by admitting air into the firebox over the fuel by numerous air-tubes, regulated by slides, through the front, sides, and back. On the Oxford, Worcester, and Wolverhampton Railway, a system was introduced by Mr. Edward Wilson, in 1858, in which air is conducted from the front of the smokebox, through several of the lower flue-tubes, into the firebox. On the northern division of the London and North-Western Railway, Mr. Ramsbottom, in 1859, introduced air under a brick-arch by two openings, 7 inches square, through the front of the firebox. (See Plates I. and II.)

With respect to the class of contrivances acting by deflection, the initiative appears to have been taken on the Birkenhead Railway by Mr. Douglas, who, in January, 1858, applied a deflecting-plate, fixed to the inner side of the door, through openings in which air was admitted, whence the air passed, and was deflected towards the fuel. In June, 1858, on the same line, a movable deflecting-plate, and underhung door, with a sector, and notches to regulate the opening for air, was applied, and more recently a plain inverted scoop or shovel. In July, 1858, on the East Lancashire Railway, Messrs. Lees & Jacques fixed the deflector to the door which was underhung, and was provided with a valve, the opening of which was regulated by a sector. They built in addition a narrow brick-arch against the tube-plate, so that the smoke and air might, by the combined action of the arch and the door, be better intermixed. In December, 1858, Mr. Sinclair, on the Eastern Counties Railway, applied a baffle-plate inside the firebox, hung over the doorway, and an underhung door, regulated by notches, with two steam-roses, one on each side of the firebox, to project steam downwards upon the fuel when the blast was off. This system was introduced by Mr. Frodsham. In 1858-9, an inverted shovel or scoop, similar to Mr. Douglas's, but longer, was, and has since been, extensively introduced, being placed in the doorway, and inclined towards the centre of the fire, in order to deliver the air directly on the top of the fuel.

It has not been attempted in the foregoing notice to describe all the plans that have been designed for the purpose of consuming coal without smoke in locomotives; the intention has been to show generally the directions in which engineers have operated for the means of insuring complete combustion. There is one feature of considerable importance which is common to all—the steam blow-pipe, or auxiliary jet, in the chimney. This is made use of for the purpose of continuing the draught of the furnace when the powerful artificial stimulus of the blast is at intervals suspended.

The necessity for admitting a supply of air is understood and acknowledged by all, whether it is introduced through the grate or otherwise. When the grate is very large, and a thin fire is maintained, the supply of air may be taken almost entirely through the grate. It is, however, indispensable to this mode of managing the fire, that the fuel should be carefully supplied at short intervals, and in small quantities, uniformly distributed,—in recognition of the principle, that the more nearly uniform the supply of fuel, the more so is the generation of combustible gases, and the more probable, accordingly, is it that the combustion of these gases and the prevention of smoke will be completed. With inclined grates, of course, the fuel is only deposited near the entrance, and it finds its way down by gravitation; and their greater ease of management in this respect, notwithstanding the frequent stoking they require, compared with ordinary level grates, mainly constitutes their superiority. With small or moderate-sized level grates, a coal fire demands still greater care than with larger grates; in the latter, smoke is to be prevented by air from the grate; but in the former the firing must be more frequent, and as the door, of course, is more frequently opened, another evil is incurred—the admission of a large proportion of cold air into the firebox, most of which passes off unburnt through the tubes, chilling the boiler and checking the production of steam. Such evils are reduced in magnitude by the simple process of admitting fresh air for the combustion of the gases otherwise than through the fuel, in greater or less proportions accordingly as it may be utilized; and as this independent admission of air is a very simple process, there can be no reason for endeavouring to dispense with it. Length of run promotes the mixture of air and gases; and in Mr. Beattie's boiler the mixture is further advanced by the interposition of fire-tiles. The brick-arch in the common firebox operates beneficially the same way, doubling the run and promoting the mixture. The baffle-plate, the hood, and the scoop operate reversely, and with considerable success, by inverting the draught of the air admitted through the doorway, and directing it over the surface of the fuel. The baffle-plate is objectionable in inducing, by its action, the suction of particles of coal through the flue-tubes, and the burning of the smokebox, unless counteracted by the internal brick-arch; and the system requires a powerful blow-pipe in the chimney. To insure the prompt and effective mixture of the air and gases, the air should be admitted and distributed at the surface of the fuel. But the ample admission and intimate intermixture of air with the gases above the fuel must be effected in conjunction with a sufficiently high temperature; and this condition has led to the adoption of various modes of heating the air or the gases, or both, prior to or at the time of mixture. Mr. M'Connell suggested the use of an air-chamber within the smokebox, in which air was designed to be heated, and led into the firebox through tubes. Mr. Wilson conducts air from the front through the smokebox and the boiler by a number of the flue-tubes; and these also partially heat the air on its way to the firebox, cooling the water in a proportionate degree. Mr. John Gray established a coke-fire on a supplementary grate, as a reservoir of heat within the fire-chamber; and Mr. Beattie embodies his reservoir in masses of fire-brick. Others attempt to heat the air by conducting it partially over the surface of the firebox shell. Expedients of this class operate in some degree advantageously; but in the adoption of such supplementary contrivances, it appears to have been overlooked that there is abundance of heat within the incandescent fuel itself to supply the demand for temperature. The problem appears to resolve itself into the immediate and thorough intermixture of a plentiful but regulated supply of air, with the ascending smoke or combustible gases, at or near the surface of the fuel. Practically, it is found necessary for this object to operate from both the front and the back of the firebox; admitting air, or using arches, deflectors, or bafflers, or doing both, as has already been described. The writer has found by experience, and in support of Mr. Hall's principle, that to burn smoke when the engine is running, it is sufficient that the air should be admitted at or near the surface of the fuel, by air-tubes distributed over the width of the firebox in the front and the back, without the aid of internal arches of deflectors. The draughts of air, through the front tubes particularly, are very strong when the engine runs ahead, and they carry the currents into the middle of the firebox, where they meet the counter-currents from the back, effect the mixture of the air and smoke, and prevent the suction of small coal through the tubes. But in all systems applied to ordinary fire-boxes, operating by means of the draughts available in a locomotive engine, aided, when the blast is off, by the steam-jet in the chimney, range of power is wanting to overtake the extremes of intense ignition and rapid generation of smoke-making gases, immediately after the steam is shut off, or when fresh fuel is added; and also to suit itself to the quiet state of the fire when the glow and excitement subside, as well as to all the varying conditions of a locomotive furnace. The means of extending the range, volume, and power of the air-currents, and of adjusting them to the wants of the furnace, are supplied by the instrumentality of the jets of steam, employed by the writer, as a means of inducting and accelerating currents of air, as applied to the engines of the Great North of Scotland Railway (Plates XLI. and XLII.), and previously noticed at page 274. The steam-nozzles, with the air-tubes towards which they are pointed, are like so many miniature blast-pipes and chimneys turned into the firebox; and they possess, relatively, the same power of urging and creating draught. By this method of steam-induction, the air-currents are delivered with such precision and velocity as to sweep the whole surface of the fuel, and forcibly to distribute the air amongst the gases. Of the virtue of forcible impingement in promoting and accelerating combustion, numerous examples occur in ordinary practice. It is in virtue of this principle that the ordinary bellows accelerate ignition; and if the upper front of a fire-place is closed with a sheet of paper, to direct the draught through the bars upon the fuel and combustible gases, the air-currents so deflected and forcibly directed upon them consume the gases, and prevent smoke. The contracted neck of the glass tube of a moderator lamp, adjusted so as to direct the ascending air within the tube upon the upper part of the flame, increases its volume and brightness. In Mr. Beattie's boiler the smoke-consuming process is, to a great extent, based on the same principle of forcible mingling of the air and gases amongst each other, as they roll against the fire-tiles and diaphragms. By the writer's process, the entire operation is consummated within the four walls of an ordinary firebox, the inducted currents sweeping up the gases and forcibly intermixing with them.

Reverting to the paper read by the writer in 1860, at the Institution of Civil Engineers, which contained the result of his observations on coal-burning in locomotive engines, during the period from 1854 to 1860, it appeared from that paper that, of the systems of coal-burning engrafted on the common firebox, Mr. Yarrow's consumed

PERFORMANCES OF COAL-BURNING LOCOMOTIVES, 1860.—WITH PASSENGER TRAINS.

System.	Designers.	Railway.	Number of Vehicles.	Gross Weight of Engine, Tender, and Train.	Average Distance run between stopping Stations.	Average Speed, including Stoppages.	Gross Consumption of Coal.			Calculated Equivalent Speed without stopping.
							Per Train Mile.	Per Mile per Ton Gross.		
				Tons.	Miles.	Miles per Hour.	Lbs. of Coal.	Lbs.		Miles per Hour.
Long firebox and combustion-chamber, . .	Mr. M'Connell	London and North-Western	13·2	115	5	27·1	35·6 Hawksbury	·31	...	42
Mid-feathers, combustion-chambers, . . .	Mr. Beattie	London and South-Western	11	102	15·7	29·6	21·0 Griff	·235	...	34
Long firebox and inclined grate, ...	Mr. Cudworth	South-Eastern	13·2	116	11	26·8	26·0 Coking Coal	·225	...	32
ORDINARY FIREBOXES.										
Door-deflector, ...	Mr. Douglas	Birkenhead	8	say 90	2·8	19·1	28·45 Welsh	·32	...	29
Brick-arch and air-bars, ...	Mr. Yarrow	Scottish North-Eastern	8	say 86	5·9	25·7	27·2 Scotch	·32	...	36·2
Brick-arch and door-deflector, ...	Mr. Conner	Caledonian	7·5	say 88	4·5	22·2	22·6 Scotch	·26	...	33·3
Door-deflector and steam in firebox, . .	Mr. Frodsham	Eastern Counties	8·7	93	5·3	21	23·3 Staveley	·25	...	29
Plain firebox and steam induction, . . .	Mr. Clark	Great North of Scotland	10·2	109·6	3·3	19	21·0 Scotch	·19	...	32
WITH GOODS TRAINS.										
ORDINARY FIREBOXES.										
Door-deflector, ...	Mr. Douglas	Birkenhead	32·	say 445	...	...	60·3 Welsh	·135	...	...
Brick-arch and door-deflector, ...	Mr. Conner	Caledonian	28·	225	6·8	11·9	37·3 Scotch	·147	...	...
Door-deflector and steam in firebox, . . .	Mr. Frodsham	Eastern Counties	26·6	228	10·2	14·5	41·5 Staveley	·182	...	...
Plain firebox and steam induction, . . .	Mr. Clark	Great North of Scotland	14·3	148	4	16·4	24·0 Scotch	·16	Passenger and Goods	...
			18·8	186	3·4	17·3	27·6 Scotch	·148	Do. do. six months, January, 1860	...
			27·	305	4	14·3	39·6 Scotch	·13	Goods only	...
AVERAGE OF PASSENGER AND GOODS TRAINS.										
	Mr. Douglas	Birkenhead	...	...	...	...	...	...	...	...
	Mr. Conner	Caledonian	17	162	5·3	16	29·5	·182	...	...
	Mr. Frodsham	Eastern Counties	17·4	161	6	18·8	32·4	·2	...	...
	Mr. Clark	Great North of Scotland	18·8	186	3·4	17·3	27·6	·148	...	...

26·8 lbs. of Scotch coal, against 22·1 lbs. of coke per mile on the same duty, showing an excess of 4·7 lbs., or 21 per cent., more coal than coke. Mr. Jenkins used 30·35 lbs. of coal against 32·43 lbs. of coke, or 6 per cent. less coal, on the Lancashire and Yorkshire Railway. When tried on the Brighton Railway, his system used about 23 per cent. more coal than coke. Mr. Lee's system, on the same line, used about 20 per cent. more coal; and the writer's system, on the same line, and on the same engine as that upon which Mr. Jenkins's plan was tried, used about 7½ per cent. more coal than coke.

On the Eastern Counties line the writer's system, working against a strong wind, used 11 per cent. more coal than coke, working in a calm; and on the North London Railway the same weight of coal as of coke was used—viz., 26 lbs. per mile.

On the South-Eastern Railway, by the writer's system, 14·4 lbs. of coking coal were used against 16·28 lbs. of coke per mile in a similar engine, on the same duty, for twelve months—showing a consumption of 11½ per cent. less coal than coke. On the Great North of Scotland Railway, engine No. 5 was put upon trial for several weeks, burning English coke alone against different kinds of Scotch coal. The trains as well as the weather were variable; but under similar circumstances, 9 per cent. less coal than coke was consumed per ton of the gross moved weight of the train. At the same time, whilst 8¾ lbs. of water appeared to be evaporated per lb. of coke, the water raised per lb. of coal averaged only 8 lbs.; thus, with a lower evaporative duty, there was a higher working duty with coal than with coke. This may be ascribed to the great command of steam afforded by the use of coal in these engines.

In comparing the coal-burning in practice on different lines, the inferior strength of Scotch coal to that of English coke and coal is to be borne in mind. The frequency of the stoppages is another element, as every stoppage of a passenger train, under ordinary circumstances, occasions a delay of from three minutes to five minutes as compared with a through running train. The annexed table contains a selection of cases from the writer's paper already referred to, in which the performances tabulated at page 217, *ante*, are repeated in abstract. It gives the average number of vehicles drawn, and the gross weight of the engine, tender, and train; the average distance run between the stopping stations; the average speed, including stoppages; the gross consumption of coal per train mile, and per ton gross per mile; also in the last column, the equivalent speeds without stoppages, or such as would have been arrived at had there been no stoppages, estimated in terms of an allowance or deduction of time of four minutes for each stoppage actually made by the passenger trains; except for the Birkenhead trains, with which, owing to the frequency of the stoppages, the actual delay is only three minutes per stoppage.

In conclusion, it should be noted that the steam should always be kept up while the engine is running. In most of the systems in use, a considerable quantity of cold air is drawn into the flue-tubes, the strength of the draught towards the flue-tubes not allowing it to approach sufficiently near the surface of the fire to mix with the smoke. In Mr. Conner's system, although the inclined plate or deflector at the doorway gives the air a downward direction, yet, when it is under the action of strong blast, the current undulates over the arch. In Mr. Douglas's and Mr. Frodsham's systems, the diverting action of the draught is still more injurious, because there is no assistance from a brick-arch in bringing together the smoke and the air: the air does not strike upon the fire, but goes at once towards the tubes. On that account, the brick-arch is a very useful adjunct in connection with the inclined entrance of the air into the box. In the plan of Mr. Ramsbottom, the air is introduced in too large a body, passing through only two places; and therefore a portion of it must pass through unmixed. The air, where it enters the box, should be introduced at the proper level for immediate mixture with the smoke.

But if it is necessary to keep up the steam when the engine is working, it is also important to keep down the steam whilst the engine is standing at the stations, or running without steam on. The chimney-blower is required, in order to create a sufficient current of air in the firebox to consume the smoke. Where tubes are placed in the front and back of the box, and induction-jets are used instead of a jet in the chimney, as the nozzles operate above the level of the fuel, the tendency is to prevent the draught through the grate, and thus to suspend the generation of smoke, as well as the consumption of fuel. For this reason, the steam is not so liable, in this system, to rise to a dangerous pressure.

Another important point is the question of management. It is desirable that any system employed should be capable of being easily managed; and as the difficulty principally occurs at high speeds with express trains, that which requires the least fuel and the least labour from the stoker is the best. With deflectors, there is a large quantity of cold air introduced into the firebox. In Mr. Conner's plan, for example, the sectional area of the entrance is 67½ square inches, or 5 square inches per square foot of grate. In the writer's apparatus, the maximum area is 4 inches per foot of grate.

In 1860, the Midland Railway Company introduced a brick-arch into the firebox, in connection with a long deflection-plate from the doorway and sliding-doors on the outside of the firebox, consisting of two half door-plates connected to slide simultaneously, together or apart, and thus to regulate the supply of air by the doorway. Mr. Markham, lately of Derby, in a valuable paper on the subject, stated that he obtained the most satisfactory results with a deflector-plate 3 feet in length, and a doorway 18 inches by 11 inches, giving 1·1 square foot of area for a grate 14 square feet area; or 11·3 square inches of air-entrance for each square foot of grate. This is largely in excess (about three times as much) of what the writer has found necessary on his system, and it signifies that a large surplus of cold air passes into the firebox besides what is actually consumed, which appears to explain why Derbyshire hard coals evaporated 6·7 lbs. of water per lb. of coal with his apparatus, whilst Durham coke evaporated 7·9 lbs. per lb. of coke, or 18 per cent. more water; whereas, on the writer's system, it has been found that coal, with 4 square inches of air-entrance per foot of grate, generally evaporated more water than was evaporated with coke.

CHAPTER XXXI.

CARRIAGES AND WAGGONS—GENERAL REMARKS.

THE carriage-stock comprises all vehicles concerned in the conveyance of passengers and their luggage, and of private carriages and horses, the vehicles to carry the latter usually being attached to, and run as part of, the passenger-train in which their owners are conveyed. The waggon-stock comprises the vehicles employed in the conveyance of merchandise, minerals, and live stock. The common varieties of vehicles employed in railway traffic are as follows:—

Passenger-train Stock.—First-class carriage, second-class carriage, third-class carriage, composite carriage, luggage brake-van, horse-box, carriage-truck. To these may be added the mail carriage, or travelling post-office, and the mail van.

Goods-train Stock.—Platform waggon, open or box waggon, high-sided round-end waggon, coke-waggon, brake-van.

Besides these, there are other waggons specially designed for special traffic, as gunpowder, salt, and lime. Also ballast-waggons, for the private use of the engineer's department.

The classification of carriages is designed to meet the various requirements of the travelling public: some preferring seclusion, ease, luxury, high speed; others preferring society, if tolerable, and economy, with moderate comfort and moderate speed; others looking to economy simply. For the sake of uniformity externally, and in many of the details, carriages are usually made of the same external length, width, and height. The underworks of the stock may thus be identical, in construction, and a uniformity of working and wearing parts is thus secured, which is conducive to economy of maintenance.

The waggon-stock should be as nearly uniform as possible. Uniformity of waggons is more important than that of carriages, as their total number and cost are much greater, and the supervision with which they are favoured is less minute; besides, the cost of maintenance is less than where many varieties of waggons exist on the same line. But whatever may be the upper works, the under works of the whole of the waggon-stock should be uniform. One of the greatest engineering evils that have been inflicted upon railway companies has arisen from the want of arrangement or consultation between the officers of different lines, in order to consider the question, common to all, as to the best plan and construction of vehicles to be used by them. The result has been, that several companies have built classes of stock unsuited to work conjointly the traffic of their own and other lines. The diversity of practice is, no doubt, partly occasioned by the growing wants of traffic, and the gradual increase in capacity and tonnage of the carrying stock, incurring unavoidable structural alterations of plan. Amongst the early contributors to the production of the railway waggon were to be found the great carriers, their agents, road-contractors, farmers, builders, wheelwrights, salesmen, graziers, timber merchants, and others whose occupations and opinions, it may be imagined, gave birth to a wide diversity of practice.

Another reason for the want of harmony in past practice has been the separation of the duties of the engine and of the carriage and waggon superintendence. The carriage superintendent, aiming at the utmost economy of maintenance in his department, has been continually adding to the quantity and weight of material employed in the construction of the carrying-stock, as the remedy for the observed failure of weak parts; and thus the stock, particularly waggons, has been increased in strength rather by adding to the mass of matter than by studying to throw the same weight of timber and iron into superior combinations. Doubtless the stock was made very lasting and serviceable; but meantime the heavy trains, handed over to the locomotive department, induced similarly the construction of heavier and more powerful locomotives, when the maximum was quickly reached, and strongly evinced by the sufferings of the permanent way.

It was of course unavoidable, from the need of enlarged dimensions and capacity, and increased strength, both in carriages and waggons, that the dead weight should be to some extent increased. The early first-class carriages weighed 3¼ tons—the bodies or upper parts being 15 feet long, 6 feet 6 inches wide, and 4 feet 9 inches high, in three compartments, to hold six passengers each, or eighteen in all. Railway carriages have gradually since been increased in weight to 5½ tons, and 5¾ tons; and in dimension of body to 20 or 21 feet in length, 7 feet 6 inches or 8 feet wide, and 6 feet 5 inches high,—outside measures for the narrow gauge. The capacity has been increased from 462 cubic feet, in the early carriages, to 1,000 cubic feet. The same increase of dimensions has taken place in the other classes of carriages. The great increase in size and weight of carriages has arisen very much in compliance with the demands of the public for greater convenience, speed, and safety. It was found that the old carriages suffered the most in cases of collision, and it became essential, with the increase of speed and length of trains, to add very much to the size, strength, and weight of the carriages. The wheels weighed originally 17 cwts.; they now weigh from 25 to 35 cwts.

First-class narrow-gauge carriages are commonly divided transversely into three compartments, from 6 to 7 feet long, and lined with cushions; in each compartment there are six seats, and there are in all seats for eighteen passengers. In some instances the seat-partitions or elbow rests are dispensed with, and the whole width of the compartment thrown open to receive four on each side, or twenty-four passengers in one carriage. Saloon-carriages, having no compartments, are getting into use, though slowly.

Second-class carriages are usually divided into four compartments, holding eight passengers each, or thirty-two per carriage. They have usually been finished with hard boarding, destitute of cushioning, on the nearly obsolete policy of making them uncomfortable, in the hope of inducing passengers to travel first-class instead. The London and Brighton Railway Company were the first to understand the value of a little encouragement to the public by meeting them half-way, and supplying comfortably-padded seats in the second-class.

Third-class carriages do not differ very much from second-class, except that they are thoroughly hard and square inside, older, and more homely.

Passenger brake-vans are made open inside for passengers' luggage; and are usually formed with a compartment at one end for the guard, with an elevated perch and glazed chamber about the level of the roof, to facilitate his supervision of the whole of the train. The brakes commonly used are sliding-brakes, with blocks of timber applied to the four wheels, brought up by a screw and handle. The weight of the brake-van is from 5 to 6 tons. Horse-boxes are constructed to carry three horses. Nearly all narrow-gauge carrying-stock is placed on four wheels.

The original form of goods-waggons generally employed for some years after the opening of the Liverpool and Manchester Railway, in 1829, was simply a platform about 10 feet long, on four wheels, with sides varying from 4 to 10 inches in height. Many of these waggons were recently employed for the transport of weighty rough goods; they weighed 2½ to 3½ tons, and carried about 2 tons of goods. The general unfitness of this style of waggons led to the adoption of portable sides and ends, which consisted of open crib-rails dropped into staples; and to these were added the costly tarpaulin or sheet to cover the goods, and bind them down (see Fig. 229). The waggon, thus appointed, made 13 or 14 feet in length, and weighed about 3½ tons. It was fitted to carry from 4 to 5 tons of goods. Crib-rails and tarpaulins have, however, been superseded to a great extent by built covered waggons, from 14 to 16 feet long, with sliding doors and movable roofs (see Fig. 230); so that, by means of the crane, a bale of goods, however heavy, can be deposited at, or removed from, any part of the interior of the waggon; and the goods may be perfectly inclosed, and protected from damage by fire, wind, or rain. Covered waggons weigh from 4 to 5 tons, and can carry, according to their dimensions, 6 to 8 tons. The cost of maintenance of ordinary open waggons is said to amount to from 7 to 10 per cent. of the first cost, as against that of the covered waggons, which is stated to be only 4 per cent.

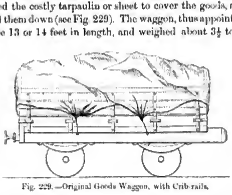


Fig. 229.—Original Goods Waggon, with Crib-rails.

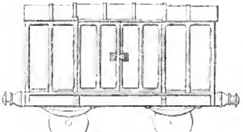


Fig. 230.—Covered Waggon.

It may be stated generally, that properly-made open waggons may carry twice their own weight of goods, and covered waggons 1½ times their own weight of goods; but that ill-designed heavily-made waggons may carry no more than their own weight in goods. The great demand for weight in waggons arose, as much as from anything else, from the absence of spring buffers at the ends, and the imperative need of strength to resist the inevitable hard concussions to which they were subject in daily use. To reduce the violence of such contingencies, and also not to add very much to the cost of the waggon, buffing-springs were applied at one end of each waggon, leaving the other end "dead," as shown in Fig. 229. The benefits of this arrangement, though partial, became evident; and as at the same time several forms of external buffers—compact, efficient, cheap, and easily applied—had been matured, and had become regular articles of manufacture, the practice of springing one end of the waggon has been gradually extended in new stock to both ends. Waggons, as formerly made, were, in long trains, likewise subjected to violent shocks in starting into motion at stations and otherwise; and therefore the draw-bars also were placed upon springs. Some persons have gone further, and placed the guard or side chains upon springs. Thus the waggon has come to be defended by springs at all points. The substitution of elastic action for dead shocks, and the use of the continuous draw-rod, has proved very beneficial in promoting the durability of waggon-stock; and there is no doubt that the extra cost so incurred has been amply compensated in saving of materials and in durability. Spiral springs for buffing and drawing, made of round or of oval steel, have answered very well.

Broad-gauge waggons have been constructed sufficiently strong to carry 20 tons of load. But they are not generally made to carry more than 10 tons. They are carried on six wheels.

CHAPTER XXXII.

CARRIAGE STOCK—EXAMPLES OF CARRIAGES FOR THE NEW SOUTH WALES RAILWAYS, &c.

1. First-class carriage for the Great Eastern Railway. Composite carriage. Underframe. Plate XLVIII.

2. First-class carriage for the New South Wales Railways. Plate XLIX.

3 Second-class carriage for the New South Wales Railways. Plate L.

4. Sectional views and underframe of first and second-class carriages for the New South Wales Railways. Plate LI.

5. Ordinary passenger brake-van. Horsebox and carriage-truck for the Australian Railways. Plate LII.

Underframes of Carriages.—The underframe is the foundation of a carriage, and should be simple, strong, and rigid, so that the body may be carried, and shocks of all kinds resisted, without deterioration or alteration of form. The alterations of form to which underframes are liable are,—hogging, or rising in the middle, and drooping at the ends, and horizontal derangement. Hogging is partially caused by the frame being unequally supported by the springs; as, for example, when the centres of the axles are too near, leaving the ends insufficiently supported. The tension-plates over the springs, if overstrained, increase the hogging strain. Horizontal derangements of the frame are caused by concussion, and also by tractive force, when exerted through the medium of the frame itself.

The distorsion of the frame vertically is checked or altogether prevented by placing the axles sufficiently apart, at a distance considerably greater than half the length of the frame, and by stiffening the side-soles, if of timber, by iron plates. Horizontally, distorsion is checked by suitably applying diagonal framing, and a firm union of the parts; by the application of buffing and drawing springs, and by the use of continuous draw-rods, in order that the pull of the train may be transmitted from vehicle to vehicle without delivering the pulling strain upon the underframe itself. But the best way to be delivered from the evils of distorsion, and the consequent cost of maintenance, is to construct underframes of iron.

The Saltley carriage-underframe (Plate XLVIII.) is a fair type of the past, and even now prevailing practice in underframes for carriages:—Of timber, having two outside longitudinal sole-bars, two headstocks or end cross-bars, two intermediate transoms or cross-bars, which receive the pull of the draw-springs; and two other intermediates to stay the diagonals, and assist in carrying the floor; lastly, four diagonal bars, to stiffen the frame horizontally, and to assist in carrying the floor; together with two longitudinal struts under the draw-bars, and a cradle to carry the springs between the middle transoms. The frame is 19 feet 11 inches long, or 1 inch shorter than the body, to prevent the lodgment of rain, and 6 feet 10 inches wide. The side sole-bars are 4 inches thick, the headstocks 4½ inches, and the middle transoms 3½ inches thick, and all of them 11 inches in depth. The diagonals are 3 inches thick, whilst the intermediate transoms are 3 inches by 9 inches deep. The cradle between the two middle transoms, to carry the buffing-springs, is 12 inches broad. The frame is bound transversely by ⅝-inch tie-rods, at each transom, so that the strut and the tie bind each other; and by wrought-iron knees at each corner, uniting the sides and diagonals to the headstocks. Further, the diagonals and struts are bound to the middle transoms with wrought-iron plates, applied below, and bolts and nuts. The struts are bound in the same way to the end beams. The buffer-springs, which are also the draw-springs, are two in number, laminated, sufficiently long to span the buffer-rods, and placed together in the central compartment of the underframe. They are confined to the cradle by bar-iron guides, between which the shackles are free to slide, and the ends bear upon cast-iron shoes cottered to the ends of the buffer-rods. The buffer-rods are 1½ inches square, except the outer portions, which are 2½ inches in diameter,—a dimension, by the way, which has never been altered since buffer-rods were introduced. They work through cast-iron slides bolted to the headstocks, and have 9½ inches of action. The draw-rods are pinned to the shackles of the springs, and are finished with hooks on the outside, where they are carried by cast-iron plates bolted on the headstocks. The axle-guards, of the **W** pattern, are bolted to the side-soles on the inside; the wheels, of which there are four, are 3 feet 6 inches in diameter, and are placed at 12 feet apart to the centre of the axles, which is 2 feet more than the half-length of the frame. Washer-plates are applied on the outside to take the nuts of bolts, and the four outer corners of the frame are squared, and bound by iron knee-plates. The side or safety-chains are hung from eye-bolts, having their bearing on iron washer-plates common to them and the longitudinal tie-bolts, near to which they are placed; so that in the event of any pull upon the side-chains, the strain is transmitted through the tie-rods to the middle transoms.

Besides the general objection to the use of timber in the construction of underframes, the frame now described is open also to the radical objection that the traction-strain—not only that exerted in pulling the vehicle itself, but in addition the strain of pulling all the vehicles following it in a train—is transmitted through the springs and thrown entirely upon the frame. The framing of every carriage must therefore be made strong enough to pull after it the heaviest train that may ever be required. This objection is obviated in the underframe of the Great Eastern Railway carriage.

Turning to the underframe of the first-class carriage of the Great Eastern Railway (Plate XLVIII.), it differs in several respects from the Saltley frame. Dispensing with the usual heavy buffing and draw-springs in the middle of the frame, a light india-rubber draw-spring is used, placed upon a continuous draw-rod. The rod, being continuous, or unbroken from hook to hook, relieves the frame from the pull of any part of the train, besides the carriage itself of which the frame forms a part; for this reason, a much lighter and cheaper draw-spring is sufficient, than where, as in the Saltley frame, the whole of the pull of the train is borne by the springs. In this frame, too, there is a remarkable absence of iron knees, except at the four outer corners, where they are applied to unite the headstocks to the side-soles. Inside, blocks of wood are substituted for knee-plates, fitted between the side-soles and the diagonals, the whole being united by bolts and nuts through the outer knees, and thus fitted to bear buffing-strains on the frame. For buffing, four external buffing-springs, in cast-iron cases, are fixed to the headstocks. The diagonals and longitudinal struts are checked and mortised into the transoms and headstocks; and, by a suitable disposition of tie-rods, every section of the frame is firmly bound. The ends of the tie-rods, it may be remarked, are increased in size, that they may be screwed without reducing the available section of the rod. This is good. The frame is 20 feet 11 inches long, or 1 inch shorter than the body. The axle-guards are of the W pattern, which is now universally employed, as it combines lightness with strength and stiffness longitudinally, and elasticity laterally. They are constructed of flat iron ⅜-inch in thickness, 3½ inches wide in the "legs," or properly the body of the guard, and 3 inches wide in the "wings." The wings are expanded so as to extend about 5 feet along the frame from tip to tip—thus abundant leverage is obtained to stay the axle-guard. The axles are placed at 12 feet apart, the same distance as in the Saltley frame. The "scroll-irons," so called from being originally of a scroll pattern, for receiving the links of the ends of the carrying-springs, are bedded on wood packing and bolted to the under-side of the frame; those at the extreme ends being turned up to take a bearing on the headstock, and thus to enable them better to resist the horizontal strain of the springs. The old-fashioned tie-rods frequently to be found uniting the lower ends of the axle-guards are very properly dispensed with, as they are not nearly stiff enough to be of any service, and, when damaged, they derange the axle-guards.

The iron underframes of the carriages of the New South Wales Railway, designed by Mr. Fowler (Plate LI.), are excellent examples of iron framing for long carriages. The total length of the underframe is 35 feet, being 1 inch shorter at each end than the body, which is 35 feet 2 inches long. The width is 7 feet. With so great a length of frame, in proportion to the width, it was deemed unnecessary to apply diagonals for the purpose of stiffening the frame, as the frame is, independently of diagonals, rigidly braced with two headstocks and seven intermediate transverse beams. This is a wise departure from the ruts of routine; more particularly since, in addition to the transverse beams, there are two intermediate longitudinal beams, reaching from headstock to headstock, partly to stiffen the cross beams and to carry the floor, and partly to afford the necessary foundation for the radial axles and for the buffing-gear. For so great a length as 35 feet, the frame is carried on four axles and eight wheels; the axles being placed at 7 feet 10 inches apart from centre to centre, on a wheel-base 23 feet 6 inches in length, or two-thirds the length of the frame. The two intermediate axles are in fixed axle-guards, and parallel to each other, constituting a fixed wheel-base of 7 feet 10 inches in length. The extreme axles are movable on fixed centres in the underframe, and are guided in axle-boxes, which are carried in triangular radial frames, of which the radius measured from the pivot to the centre line of the axle is 5 feet 2⅔ inches, with which the extreme axle works truly radial on curves of any radius. This precise length of radius is the result of an ingenious geometrical investigation made by Mr. J. D. Baldry, under the instructions of Mr. Fowler. The conclusion may be expressed algebraically by the following general formula for finding the proper length of radius, which is applicable also to vehicles of any other proportions:—

$$x = \frac{a - \frac{b^2}{a}}{2};$$

in which $a =$ the sum of half the fixed wheel-base and the distance of the radial axle from the fixed base; $b =$ half the fixed wheel-base; and $x =$ the length of the radius for the radial axle, such that the axle works truly radial to curves of any radius.

In the present example, $a = 3{\cdot}9166 + 7{\cdot}8333 = 11{\cdot}75$ feet; and $b = 3{\cdot}9166$ feet; and the radius, $x = \dfrac{11{\cdot}75 - \frac{3{\cdot}9166^2}{11{\cdot}75}}{2}$ $= 5{\cdot}223$ feet, or 5 feet 2⅔ inches. With so short a fixed wheel-base—less than 8 feet—these long carriages are adapted to pass with facility along curves of any radius; and there cannot be any reasonable doubt that they typify the carriage of the future.

With respect to the provisions for drawing and buffing, the gear is compactly placed at each end of the underframe. The draw-rod is not continuous, but is limited at each end to a volute draw-spring, which is applied to the breast beam or plate. The pull is, however, safely transmitted through the two intermediate longitudinal beams of the underframe, and thus an equivalent for a continuous draw-rod is provided. At the same time, the design is open to the objection that the pull is taken from the extremities of the frame; and the obliquity of the pull on curves, for so long a frame, would have been somewhat serious, but for the radial freedom of the extreme axles. The buffing-gear is skilfully designed with a compensating beam spanning the buffing spindles, by means of which the pressures on the inner and outer buffers are equalized on curved portions of the line. The springs consist of a series of india-rubber rings, 5 inches in diameter, on each spindle; and the slack of the spindles is taken up by light helical springs on the ends of them.

As to the scantling of the framing, the side beams are of L section, 9 inches in depth by 4½ inches wide on the flange, which is uppermost, by ⅝-inch thick, stiffened by angle iron of the same thickness, 4½ inches wide by 3 inches deep, applied laterally. The ends consist of a ⅝-inch plate, 9 inches deep, stiffened with channel iron of the same depth, and 5 inches wide on the upper and lower flanges. The frame is, for convenience of transport, constructed in two halves, joined at the mid-length, and the middle ends are of L iron, 9 by 3 inches, and ⅝-inch thick. The other principal members of the frame are of T iron, 8 inches deep by 5 inches wide, and ⅝-inch thick, the flanges being in all cases uppermost, to afford direct support for the floor. The wheels are of solid wrought iron, 3 feet in diameter, with eight double spokes, 3½ inches wide by 13/16-inch thick at the nave, tapered to 11/16-inch thick at the rim. The rim is 1 inch thick, and 3½ inches broad, turned on the outer surface. The nave is 7 inches long and 8 inches in diameter, or 1¾ inches thick round the axle. The tyre is 2 inches thick at the middle of the tread, and 5 inches broad; the slope of the tyre is 1 in 20. The flange is 1 inch high, and in section is formed with curves of 1 inch radius. The tyre is fixed to the wheel with four rivets, ⅝-inch in diameter. The axle is 4 inches in diameter at the middle, 4¾ inches at the back of the nave of the wheel, and 4½ inches within the nave. The wheels are placed on the axle at a distance apart of 4 feet 5⅜ inches between the backs of the tyres. The journals or bearings of the axle are 3¼ inches in diameter by 8 inches long, and are 6 feet 9½ inches apart to centres. Each wheel is fixed on the axle with a steel key ⅝-inch broad and ¾-inch thick.

I.—First-class carriage for the Great Eastern Railway. Composite carriage. Underframe. Plate XLVIII.

The body of the Great Eastern first-class carriage is 21 feet long and 8 feet wide, outside measure. It is divided into three compartments, each 7 feet long on the outside; each compartment has seats for six, making for the whole carriage eighteen passengers. The inside clear height at the middle is 6 feet 3 inches from the floor to the roof. The gross cubic capacity of the body, measured externally, is over 1,000 cubic feet, or nearly 60 cubic feet per passenger. The framing of the body is of mahogany, except the roof ribs, which are of ash. The roof is of pine boarding 1 inch thick, and the floor is of two ¾-inch thicknesses of boards crossing each other. The body rests not directly on the underframe, but on brackets faced with india-rubber, fastened to the underframe. The elastic cushions, thus interposed, soften the motion of the carriage.

The composite carriage is 8 feet 2 inches wide outside, and 18 feet 1 inch in length. It is in three compartments, of which the centre is first-class, and is 7 feet 1 inch long; the two outer compartments are second-class, and are 5 feet 6 inches long. The carriage is designed to carry eight first-class and twenty second-class passengers,—that is to say, four first-class on each side, and five second-class on each side. The gross capacity per first-class passenger is 47 cubic feet, and per second-class passenger 29 cubic feet.

The carriage-underframe has already been described, page 303.

II.—First-class carriage for the New South Wales Railway. Plate XLIX.

The body of this carriage is designed to have two ordinary first-class compartments, 7 feet 1 inch long, and a saloon between them, 21 feet long—making the total length of the body 35 feet 2 inches. The saloon has accommodation for eighteen passengers, and the end compartments for six each,—in all, thirty passengers. The body is 7 feet 9 inches wide outside, and is constructed with a double roof, of which the space between the inner and outer roof is 6 inches; from the under-side of the body to the top of the outer roof, the height is 7 feet 5 inches. The cubic capacity of the body, taken to the inner roof, is at the rate of 59 cubic feet per passenger. The whole of the framing is of Moulmein teak; the panels and doors are of mahogany.

III.—Second-class carriage for the New South Wales Railway. Plate L.

The body of this carriage is of the same length as that of the first-class,—35 feet 2 inches,—and 7 feet 9 inches wide outside. It comprises six second-class compartments, and a compartment for the guard, with the brake. Of the passenger compartments, four are double, 5 feet long, for ten passengers; and two are single, 4 feet 1 inch long, for five passengers,—in all, the carriage has accommodation for fifty passengers, with a gross capacity of 29 cubic feet per passenger. The guard's compartment, which is 8 feet 1 inch high to top of roof, outside measure, is 7 feet long. The brake is here worked by the guard, being an ordinary screw brake, with blocks applied to the four fixed wheels.

IV.—Sectional views and underframe of the first and second-class carriages for the New South Wales Railways. Plate LI.

The sectional views of these carriages do not require any further notice; and the underframe has already been described, page 303, where also is given a formula embodying the geometrical investigation for finding the length of radius of the radiating axles.

V.—Ordinary passenger brake-van. Horse-box, and carriage-truck, for the Australian Railways. Plate LII.

The brake-van is arranged in two compartments—one for the guard, with windows at the end to give a look-out, and the other for passengers' luggage, with doors at each side. A dog-box is constructed at the end of the luggage compartment. The body is 16 feet long, and 7 feet 6 inches wide. The brake is known as the "sliding-brake." The axle-boxes on each side are connected by a straight bar, on which a pair of sliding sockets are fitted, carrying the brake-blocks. They are moved by the usual intermediate shaft and a screw power.

The horse-box is constructed to carry three horses, with two dog-boxes, one at each end. The total length of the body and the underframe is 14 feet 6 inches, and the body is 7 feet 9 inches wide outside. The horse-compartment is only 10 feet 3 inches long, and has a gross capacity of 200 cubic feet for each horse. The axles are only 8 feet apart.

The carriage-truck is 14 feet 6 inches long, and 7 feet 9 inches wide, like the horse-box, with the axles at 8-feet centres.

CHAPTER XXXIII.

CARRIAGE STOCK OF THE NORTH LONDON RAILWAY—WILKIN AND CLARK'S BRAKE.

FIRST and second-class carriages and brake-van, designed by Mr. William Adams, Locomotive Superintendent, for the North London Railway. Plate LIII.

The rolling stock of the North London Railway is designed specially with a view to working the peculiar traffic of that line. The carriages are of three classes—first, second, and composite first and second—the latter of which are supplied chiefly as smoking carriages. They are lit up at night with gas, of which a reservoir is carried in the brake-van, where it is conducted by pipes over the roofs of the carriages, with a branch to each compartment. As the carriages are made up into permanent trains, and never separated in the ordinary working of the traffic, spring buffers are only applied at one end of each carriage, the other end being fitted with dead buffers; and the carriages are so arranged that spring buffers are matched with dead buffers. The arrangement is the same as that employed on the South-Eastern Railway. There are thirty gas-lights in each train, supplied with gas from two gasholders, one in each brake-van. The holders carry about 200 cubic feet of gas, which is sufficient to serve the train for from two hours to two hours and a half. The body of the first-class carriage is 27 feet 1⅜ inches long, and 8 feet wide outside, with a gross capacity of 1,520 cubic feet. There are four compartments, 6 feet 9¼ inches long each, to hold eight passengers in each, or, in all, thirty-two passengers; and the gross capacity of the body per passenger is 48 cubic feet. The carriage runs on four wheels, 3 feet 6 inches in diameter, with the axles at 15-feet centres. The weight of the carriage in running order is 7 tons 15 cwts.; and loaded, allowing 1 cwt. per passenger, 9 tons 7 cwts.

The second-class carriage is longer than the first-class, being 27 feet 11 inches long, with five compartments, each 5 feet 7 inches long. The width outside is 8 feet, to hold five passengers in the width, or, in all, fifty passengers. The gross cubic capacity of the body is 1,560 cubic feet, or 31 cubic feet per passenger. The wheels, four in number, and 3 feet 6 inches in diameter, are at 15-feet centres. The seats and backs of each compartment are comfortably cushioned. The bodies are open from end to end, above the backs of the seats, and ventilation is in general amply provided for. The weight of the second-class carriage in working order is 7 tons 18 cwts. 2 qrs.; and, allowing 1 cwt. per passenger, 10 tons 8 cwts. 2 qrs. loaded.

The brake-van is 18 feet 6 inches in length, and, like the carriages, is 8 feet wide. It has four wheels, 3 feet 6 inches diameter, at 11-feet centres. The chamber for the gasholder occupies about 6 feet of the length of the body; the remainder of the body is designed to hold luggage and to accommodate the guard, who occupies an elevated position under a raised part of the roof, which is glazed on all sides. There the brake is worked. The weight of the van in running order is 7 tons 10 cwts.

To compare with the stock of the North London Railway, it may be observed that the newest carriages of the Metropolitan Railway are constructed with bodies 20 feet long and 8 feet 3 inches wide outside, with an average gross capacity of about 1,200 cubic feet of body. The first-class are divided into three compartments, and the second-class into four, accommodating respectively forty-eight and eighty passengers. The weights of the carriages are about 7 tons 2 cwts., and 7 tons 4 cwts., each, respectively. They are specially buffered, and coupled at the centre in pairs, constituting "twins."

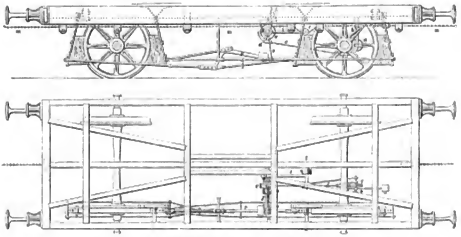
The following is a comparative statement of the relative weights, capacities, and accommodation of the North London and the Metropolitan carriages:—

Railway.	Class.	Weight.		Passengers.	Passengers per Ton.	Gross Capacity per Passenger.
		Tons.	Cwts.			Cubic Feet
North London,	1st Class.	7	15	32	4·0	48
	2nd Class.	7	18½	50	6·3	31
Metropolitan,	1st Class.	7	2	24	3·3	50
	2nd Class.	7	4	40	5·4	30

From which it appears, that whilst the North London carriages hold from 12 to 18 per cent. more passengers per ton of dead weight, they afford practically an equal gross capacity per passenger.

Particulars of the working of passenger trains on the North London Railway have already been given at page 284. To the information there given it may be added, that the maximum running speed is at the rate of about thirty-five miles per hour. The excellent performance of the engine is supplemented by the brake (Wilkin and Clark's), which is applied to every vehicle in the train, in consequence of which the train is brought up in a very short space. To each vehicle two pairs of pendulous brake-blocks are hung in the usual way. The brake is worked by a ⅜-inch chain, carried on sheaves along the centre of the train, united by coupling-hooks at each carriage. In the centre of each carriage the chain hangs down like a festoon, and passes under two pulleys attached to pulling-rods fitted to the block-hangers. When the chain is tightened, the centre pulleys are raised, and the blocks pulled on the wheels with a collective force of about 3 tons for each vehicle. When the chain is slackened, the pulleys, assisted by a back-weight, descend by gravity to their normal position, and free the brake-blocks. The chain is tightened from either end of the train by means of two transverse axles, driven by steel-faced friction-wheels 20 inches diameter, screwed by manual power against the van-wheels. The momentum of the van is thus made to retard the whole of the train; and is so powerful, that a train of eight vehicles can stop the largest engine under full steam. This system, which has worked successfully for several years on the whole of the North London Railway stock, and partially upon that of ten other railways, has the defect, that, in the event of the chain breaking, the brake is for the time useless. To remedy this defect, Messrs. Wilkin and Clark introduced, in 1869, on the Metropolitan Railway, an improved brake, wherein the strain upon the through-chain is reduced from 3 tons to the moderate load of 35 lbs.; and if the chain break by accident, the brake applies itself automatically. A $\frac{5}{16}$-inch chain is passed along the centre line of the train, united by couplings at each carriage. The guard, previous to the train starting, winds up the slack of the chain, and, by means of a friction-strap attached to the hand-drum in the van, keeps it tight while the train is running. The chain, while tight, keeps an "operating lever" under each carriage in suspension; and when slackened, the operating levers descend, and one axle upon each carriage develops the power for applying the brake upon the same carriage. By this means the moderate load of 35 lbs. on the operating lever develops a power of about 3 tons on the brake-blocks.

Having thus generally described the principle and the action of the brake, reference may be made to the illustrations (Figs. 231 and 232), showing the brake as applied to an ordinary four-wheeled carriage. The eccentric, *a*, is hung in contact with one of the axles of the carriage, having a stroke of 1½ inches; by means of the pendulous lever, *b*, and the pulling-rod, *c*, and the pall, *d*, a continuous feed-movement is imparted to the ratchet-wheel, *e*, which is loose on the shaft, *f*, but this motion is imparted only when the brake is required. The ratchet-wheel is inclosed between two friction-discs, *g*, *g*, one on each side, which are held on the shaft by two feather-keys, and revolve with it, but may slide laterally on the shaft. One of these discs is adjusted in a fixed position on the shaft, but the other is acted on by a helical spring, which is compressed, when the "operating lever," *i*, is allowed to descend by its own gravity,—the expanding action of two rollers thrusting apart, by their action on inclined planes, two collars. These collars are loose on the shaft, and are prevented from revolving. The operating lever, *i*, is furnished with a pulley, *l*, at its extremity, which receives and is raised by the small chain, *m*, passing under it, and over fixed pulleys, throughout the entire length of the train,—the chain being made in sections, coupled between every two carriages. By the lateral pressure on the friction-discs, *g*, *g*, consequent on the fall of the operating lever, they embrace between them the ratchet-wheel; and the revolving movement of the latter is communicated by this frictional contact to the shaft, *f*, and to the sheave, *n*, which winds up the chain, *o*, connected to the end of the brake-lever, *p*. The brake-lever, thus forcibly raised, applies the blocks to the wheels, until the pressure ultimately accumulated on the brake reaches the limit of the frictional adhesion between the ratchet-wheel and the discs. Then, although the ratchet-wheel continues to revolve so long as the carriage is in motion, the motion of the spindle and the discs is arrested, and they remain at rest.



Figs. 231, 232.—Continuous Train-brake, by Wilkin and Clark.

The fall of the operating lever is from 6 to 7 inches at the extremity, comprising about 3 inches for clearance and taking up slack, and 4 inches for action on the lateral wedges and tightening the friction-discs. It is obvious that the pressure with which the brake-blocks are applied remains the same, however much they may be worn.

Two important advantages are gained by the self-action of the brake. One is, that should the train accidentally break away on an embankment or in any other dangerous situation, the chain is immediately broken, and the brake at once acts on all parts of the train. This, it is said, has been frequently tried, and always with a good result. The other advantage is, that the adjustment of the maximum power of the brake is not under the control of the guard; so that the brake may be adjusted to within the skidding power, and its power cannot be increased by the guard.

The blocks are suspended from the framing; but instead of hanging hard in the brackets, the straps have elongated eyes, with a cushion of india-rubber 1¼ inch above and below the pin, to deaden the tremor.

It is said that, at a speed of forty miles an hour, trains are usually stopped in twenty-five seconds, by means of Wilkin and Clark's brake, after traversing 280 yards, in regular work. The annexed memorandum shows the results of some trials of the brake on the Midland Railway. The train consisted of two vans and six carriages, fitted with Clark's brake; also two carriages and two horse-boxes without use of brakes: twelve vehicles in all (four brakes not in use):—

MEMORANDUM OF TRIALS OF CLARK'S BRAKE, ON THE MIDLAND RAILWAY, DURING THE JOURNEY FROM BEDFORD TO ST. PANCRAS STATION, JULY 12, 1870.

Arrived at	Estimated Speed.	Brake applied, and Train came to rest.	Remarks.
	Miles per Hour.	Seconds.	
Ampthill, .	45	50	Engine brake not applied. Speed reduced to four miles per hour in thirty seconds.
Fleetwick,	42	30	Engine brake not applied.
Harlington,	43	18	Engine brake applied; ascending gradient, 1 in 200.
Leagrave, .	44	25	Do. desc. 1 in 300.
Luton, .	50	27	Do. desc. 1 in 175.
Harpenden,	53	28	Do. desc. 1 in 300.
St. Albans,	50	35	Do. desc. 1 in 300.
Radlet, .	41	35	Do. level.
Elstree, .	40	26	Engine brake taken off. Gradient ascends 1 in 400.
Mill Hill, .	45	35	Gradient desc. 1 in 176. Engine brake applied.
Hendon, .	39	28	Do. desc. 1 in 200.

On the journey from Luton to St. Pancras, there was a gain in time of six minutes.

Estimated weights:—

Engine and Tender, ...	53 tons.
Brake Train, ...	64 tons.
Extra Vehicles, ...	33 tons.
	150 tons.

A series of trials of the brake was made in April, 1870, on the Bromsgrove incline of the Midland Railway, 1 in 37, and the following are the results, as officially reported:—1. A train of six carriages and two vans, fitted with the brake, was stopped in the space of 1 mile 354 yards, having been running down the incline at a speed of forty miles an hour, when the brakes were applied. 2. The same train was then stopped in 600 yards down the incline, with an initial speed of thirty miles per hour. 3. With an initial speed of twenty-three miles per hour, the train was stopped in 460 yards. 4. An important test was then made of its self-acting properties. The train was run up the incline, and was severed from the engine while running, as if by accident. The brake stopped it immediately on the incline, without assistance from any one. About forty engineers were in the train to witness the trials.

On another occasion, a series of trials was made on the Midland Railway, with a four-coupled engine and a train of six carriages and two vans, fitted with Clark's continuous brake:—1. On a gradient of 1 in 393 down, with Clark's brake and the tender brake only. Initial speed, forty-five miles per hour; time, 28 seconds; distance, 373 yards. 2. On a gradient of 1 in 90 down, to try the effect of Clark's brake when left to its own action, in the event of the train breaking away. Vehicles detached, six carriages and one brake-van. Initial speed, thirty-three miles per hour; time, 15 seconds; distance, 200 yards. 3. On a gradient of 1 in 90 down, for the same object. Initial speed, fifty miles per hour; time, 27 seconds; distance, 330 yards. 4. On a gradient of 1 in 623 up. To try in what distance the power of the engine could be overcome, and the train brought to a stand, by the application of Clark's brake only, the engine being in full gear forward, with full steam on. Initial speed, forty-five miles per hour; time, 2 minutes; distance, 1,220 yards; steam pressure, 145 lbs. per square inch.

The London and North-Western Railway Company have (December, 1870) adopted Wilkin and Clark's brake, and are now engaged in fitting it on their express trains, with the intention of extending it to the whole of the carriage stock.

CHAPTER XXXIV.

WAGGON STOCK—EXAMPLES OF WAGGONS.

1. Covered goods-waggon, for the Australian Railways. Covered goods-waggon for the Great Southern and Western Railway. Round-ended goods-waggon for the Great Eastern Railway. Plate LV.

2. Cattle-waggon for the Ulster Railway. Eight-ton coal-waggon for the Ulster Railway. Open box-waggon for the Great North of Scotland Railway. Plate LVI.

The covered goods-waggon for the Australian Railways is constructed according to the usual English pattern. It is framed of oak, and lined with sheet-iron, sides and roof. The underframe is of the usual design, with transverse buffing and drawing springs, similar to the underframes of ordinary carriages. The corner pillars and end pillars of the body are applied direct, and fixed to the underframe on the outsides for its whole depth, whilst the intermediate side pillars are mortised into a side-rail bolted to the underframe. This method of securing the end pillars is admirably adapted for uniting the body and underframe, and enabling the former to resist the longitudinal shocks to which goods-waggons are liable when in use. The body is 15 feet long, 7 feet 6 inches wide outside, and 5 feet 10 inches high inside at the centre. The flooring is of 1½ inch thick boards. Double-sliding doors are applied to each side of the body, to open 6 feet wide. A portion of the roof is also constructed to slide transversely, so as to make an opening half the width of the roof, and 6 feet long.

The covered waggon of the Great Southern and Western Railway is a type of the covered goods-waggon in general use on the Irish Railways. It is so constructed as to be employed also to carry cattle, which suits the general traffic of the country. The sides and ends are constructed with ventilating openings, and the floor is fitted with iron draining-plates. The body lining is of wood, the floor is of 2-inch deals, and the roof is of corrugated iron, except where it is open for a width of 4 feet 4 inches. The body is 14 feet long, and 8 feet 2 inches wide, 5 feet high at the sides, and 6 feet 10 inches high at the centre. All the pillars are applied direct to and bolted to the side and end bars of the underframe. The underframe is well constructed with iron knees, external spring buffers, and a continuous draw-rod with springs.

In the round-ended goods-waggon for the Great Eastern Railway, the underframe is like that of the first-class carriage. It is fitted with external spring-buffers, and a continuous draw-rod. The body is 16 feet long by 7 feet 9 inches outside. The sides are 2 feet 3 inches high above the flooring, which is 3 inches thick. The ends are formed half-round, and are 4 feet high above the floor. The lining is 1¼ inches thick, screwed to pillars lapped on and bolted to the underframe. The wheels are 3 feet in diameter, and are placed at 9-feet centres between the axles. The side and end-soles are 11 inches deep and 4½ inches thick. The intermediate cross-bars are of the same depth, and four inches thick; and the longitudinal struts between them are 3½ inches thick, of the whole depth. The diagonals are 3½ inches thick and 7 inches deep.

The cattle-waggon for the Ulster Railway is an excellent example of combined iron and wood framing of great strength, to resist the swaying of heavy cattle. The body is 15 feet 8 inches long and 8 feet 3 inches wide outside. The sides are 6 feet high. The upper rails of the body are strongly tied longitudinally and transversely by iron rods. Altogether, this cattle-waggon is a good study.

The coal-waggon for the Ulster Railway is 15 feet 6 inches long and 8 feet wide outside. The sides are 2 feet 8 inches high from the floor, and the ends 3 feet 2 inches high. The body is firmly bound at each corner by a single angle-plate for the whole depth, and each end is fortified by two pillars lapped upon and bolted to the headstocks of the underframe. The middle portion of each side, for 5 feet in width, is hinged to the side-rail, upon which the sides are supported, to fall down as a flap when required for loading or unloading. The waggon is provided with spring-buffers at each end; but the draw-rods are not continuous, and so the underframe is made of great scantling, and is bound with iron knees and tie-rods, to enable it to support the tugs and concussions.

The open box-waggon, by Mr. Wm. Cowan, for the Great North of Scotland Railway, is a much better-designed waggon for coal, &c., than the one just noticed. The underframe is of lighter scantling, and yet is strong enough; it is firmly bound together, and is fitted with a continuous draw-rod on springs, as well as buffered with springs on the outside. The guard-chains are also mounted with springs. The body is 16 feet long by 7 feet 7¼ inches wide outside, 2 feet deep inside. The floor and sides are of 2½-inch boarding, bound at each corner by very deep angle-plates, reaching 12 inches each way. The ends are also stayed by two pillars bolted to the headstocks. The underframe is made of the unusual width (for the 4 feet 8½-inch gauge) of 6 feet 11¼ inches outside. Thus a great width of body is simply arrived at without much overhang, the side-rail being bolted directly to the sole-bars. At the same time, the axleboxes are placed more widely apart than is usual, thus affording a beneficial overhang from the wheel, and scope for some elastic action in the axle, which is favourable to durability. Plenty of room is also allowed for a long journal or bearing, and a grit shield at the back of the axlebox; whilst all the parts are open for examination.

CHAPTER XXXV.

CARRIAGE AND WAGGON STOCK FOR THE 3 FEET 6-INCH GAUGE—EXAMPLES.

Narrow-gauge second-class carriage for the Norwegian Railways. Narrow-gauge timber-waggon for the Norwegian Railways. Plate LIV.

The question of the employment of a narrower gauge for outlying districts—narrower than the standard gauge of the country—is now (1870) under discussion, with reference to the extension of railway communication in India, where the standard gauge is 5 feet 6 inches. It is contended that a narrower gauge, say 3 feet 6 inches, would satisfactorily answer the requirements of the branch districts; and there is little doubt that a narrower gauge for branch lines will ultimately be adopted. In Norway, the narrower gauge of 3 feet 6 inches has been extensively adopted, and has become, in fact, the standard gauge of the country. There are three lines of railway to this gauge:—the Hamar-Elverum, the Trondheim-Storën, and the Drammen-Randsfjord lines. The only line on the wider or 4 feet 8½-inch gauge is the Christiania-Eidswold line, which was constructed in 1854, under Mr. Robert Stephenson.

The stock selected for illustration in Plate LIV. clearly illustrates the variation of treatment, in detail, suitable for the 3 feet 6-inch gauge, as compared with the 4 feet 8½-inch gauge. It is neatly worked out by Mr. Carl Pihl, the Government Railway Engineer in Norway.

The body of the second-class carriage is 6 feet 10 inches wide, or about double the width of the gauge, and 20 feet long outside. This width is only 6 inches less than what was given to passenger carriages in England twenty years ago, whilst the length is greater. The body is divided into two compartments, in each of which there are two seats transversely, and also two seats back to back in the middle of the floor. Allowing 19 inches width of seating for each passenger, the whole carriage accommodates twenty-eight persons, for whom 820 cubic feet of gross capacity of body is provided, or 29 cubic feet per passenger, which is practically the same as is provided on the ordinary gauge. The underframe is of wood, the side-soles being 10 inches deep and 4½ inches thick; the headstocks and the middle transoms are 10 inches deep by 4 inches thick; the diagonals 4 by 6 inches. The wheels are 2 feet 6 inches in diameter, with tyres 4¾ inches wide; the axles are 3½ inches in diameter at the middle, 4 inches in the wheels, and 2¾ inches at the journals, which are 5½ inches in length. The axles are placed at 10 feet apart. As the sole-bars of the underframe considerably overhang the wheels, the axle-bearings and the bearing-springs are placed entirely within the side-soles, on which lateral brackets are bolted, to take the necessary bearing upon the springs. In the design of the underframe, the usual pair of buffers at each end is dispensed with, and a central buffer substituted and fixed on the draw-rod; but inasmuch as the draw-rod is continuous, there is no elastic buffing action. The draw-rods therefore require to be exactly coupled, without any material degree of slackness; and for this purpose there is a simple pall-latch hinged to one buffer-head, which can be dropped over a pin in the opposed buffer-head. But though the buffers are inelastic, the draw-rod pulls the carriage through the medium of a pair of volute springs at the middle, which are placed base to base on an intervening disc.

The timber-waggon is 18 feet long and 6 feet 6½ inches wide. The side-soles are properly made thinner and deeper than those of the carriage, being 3 inches by 12 inches deep, as against 4½ inches by 10 inches in the carriage,—thus securing, with one-fifth less material, equal vertical stiffness to sustain a load which is necessarily applied at the centre, in the timber waggon, as compared with a distributed load in the carriage. The detail of the underframe and the wheels is the same as for the carriage. The bearing-springs have 2 feet 10 inches span, and are 3 inches wide, and have their bearings on brackets bolted to the sole-bars of the frame. An ordinary lever-brake is applied to act on two wheels at one side of the waggon. The intelligent disposition of the short levers on the brake-shaft deserves notice. They are placed, not diametrically opposite to each other, as is usual, but obliquely, in order that the leverages exerted upon the two brake-blocks may be equal in all positions of the levers and states of wear of the blocks.

The stock of the Festiniog Railway, in Wales, merits some notice, on account of the narrow gauge, 1 foot 11½ inches, of the way. The performance of locomotives on that line has already been noticed (page 285). The passenger-carriages are 10 feet long, and 6 feet 3 inches wide outside. The floor is only 9 inches above the rails, and the carriage stands 6 feet 6 inches high above the level of the rails. Each carriage has four wheels, 18 inches in diameter, and at 4-feet centres. The body is constructed with a longitudinal partition down the middle, with seats back to back, thus throwing the weight of the passengers between the rails. Each second and third-class carriage holds ten passengers, and gives 35 cubic feet of gross capacity per passenger,—comparing in this respect advantageously with the carriages of the 4 feet 8½-inch gauge. The couplings are central, similar to the screw-couplings in common use; they are 15 inches above the rails, and work on volute springs. The side-springs, over the axles, are also volutes. The buffers are central, and are placed 4½ inches above the couplings, working against a buffing-spring of 2 feet span.

CHAPTER XXXVI.

FIXED PLANT—TURNTABLES, TRAVERSERS, HOISTS, WATER-SUPPLY, SWITCHES, AND CROSSINGS.

1. Carriage turntable. Engine and tender turntable. Plate LVII.

2. Water supply:—Water-tank, pumping machinery, and water-crane. Plate LVIII.

3. Switches and crossings:—Wild and Parson's switch, Parson's reversible steel crossing. Plate LIX.

I.—Carriage turntable. Engine and tender turntable. Plate LVII.

Turntables are divisible generally into two classes—those for turning carriages and waggons, and those for turning locomotives with or without tenders, which are necessarily much larger than the carriage turntables.

Carriage Turntables.—Originally 12 feet in diameter, they have been gradually increased in accordance with the increasing wheel-base of carriages, up to 15 and 16 feet in diameter. The turntable illustrated belongs to a very simple class, and is bedded on stone and brickwork. The table is 15 feet in diameter; it forms a complete circle, and revolves on a central pin and eight live rollers. The central casting rests on the foundation, with a square base 3 feet square; and a socket bored out to a diameter of 4 inches, to receive a wrought-iron pivot-bolt of the same size. This bolt is hollowed spherically at its upper end, to receive a cast-iron pivot upon which the central portion of the table is hung, by means of bolts and nuts, which afford a means of adjustment of level. The circular roller-path forming the outer portion of the base is cast in four segments bolted together, and turned in the lathe to the slope of the rollers; it is united to the central casting by four tie-rods, with adjustable nuts. The rollers are 12 inches in diameter and about 2¼ inches broad; they run on radial rods, bolted to a central casting, which is bored out, and turns on the central bolt; and the ends of the radial rods are connected by a wrought-iron ring behind the rollers, by means of which the rollers are steadied and kept at their proper distances with regard to each other. The table is constructed of wrought-iron, excepting only the central casting. The roller-path of the table consists of an ordinary flat-footed rail inverted, so as to present its upper table as a bearing upon the rollers. It is united to the centre by angle-iron framing, and the whole is covered by chequered wrought-iron plates, bolted or rivetted to the framework; two cross-lines of bridge-rail are fixed on the table. It is surrounded by a kerb of cast-iron, fixed to the outer foundation, into which the palls are dropped, by which the table is held in position.

The engine turntable is 40 feet in diameter, or more properly 40 feet in length. It consists of two parallel wrought-iron girders, united at the centre by a massive casting, to take the central bearing; a pair of rollers are fixed at each end, to run on a circular path. The centre and the circular path are bedded on foundations of stone and brick. The cast-iron base for supporting the central pivot is 5 feet 6 inches square, with a socket bored out to 9 inches in diameter, to receive the pivot-bolt, which is of wrought-iron, turned to the same diameter. An excellent method of distributing the load safely and equally upon the casting is here applied, by forming a collar on the pivot, which bears on the upper end of the socket; the load is thus fairly applied to the upper surface of the casting. The bolt is formed to receive a steel-die into its upper end, to take the bearing of the central pivot, which is fixed in a cast-iron disc, from which the central portion of the table is suspended by means of bolts and nuts. The main girders are of wrought-iron plates, top, bottom, and web, rivetted together; they are 3 feet 1 inch deep at the middle, and are tapered to a depth of 2 feet 3 inches at the ends. The central casting uniting the girders is of the same depth as they are; it is bored out to turn on the pivot-bolt. A line of rails is carried by the girders, on wooden longitudinal sleepers. A platform 12 feet wide, with fencing on each side, is constructed of iron plates, from end to end, at or near the level of the rails. The table is carried on a pair of rollers, 2 feet in diameter, at each end, running in brackets attached to the girders. The roller-path consists of an ordinary flat-foot rail laid on the foundation. There is no propelling power, as the engine and tender are pushed round direct by the men.

Traversers.—These are convenient and economical substitutes for turntables. The traverser is a low flat table which runs on a line of rails laid across a number of other lines of rail. It is fitted with a line of rails on itself, the rails overhanging at the sides, and at the lowest practicable level, so as just to clear the fixed lines of rail. Thus, when a carriage or waggon is run on to the back of the traverser—a process which is rendered easy by the aid of short inclined planes attached at the ends of the table—it is traversed over any number of lines of rail, and then run off and deposited on another line. The wheels on which traversers are carried are placed in pairs, one a little behind the other, to enable the traverser to pass the gaps in the transverse rails without shocks.

Hoists.—Hoists are employed at breaks of level, as, for example, between the level of the street below and the level of the rails above, for the conveyance of goods from the lower to the higher level, and *vice versa*. Their only merit is their compactness; for inclined planes are preferable to hoists, and are adopted wherever there is room for them. Waggon-hoists consist generally of a platform sufficiently large to carry a railway waggon, and suspended between four pillars or posts at the corners, which guide the platform and carry balance-weights. The 10-ton waggon-hoist employed at the Bishopsgate station of the Great Eastern Railway is framed of timber. Fixed segmental racks are bolted to the corner pillars, with which are geared four wrought-iron screws of between five and six turns, 15 inches in diameter and 3 inches in pitch. The screws traverse on four shafts, one at each corner, 4 inches in diameter; and the elevating action is obtained by the revolution of the shafts, which carry round with them the screws, the latter having feathers which slide in slots on the shafts. The screws are made right and left-handed, in pairs, to counteract each other laterally, and destroy any tendency to twist the platform. The upright shafts are driven by means of a system of horizontal shafts and mitre gearing, having teeth of 2 inches pitch and 5 inches broad, worked by a twenty horse-power steam-engine. The balance-weights, one at each corner, are of cast-iron, and work within enclosures formed by sheet-iron sheaths bolted to the pillars. The weights are connected by ⅝-inch chains, passing over covered pulleys at the top to the cast-iron beams carrying the platform. Reversing gear is applied between the engine and the machine, that the platform may be raised or lowered as required. The platform is 21 feet by 10 feet, having four longitudinal timbers, two of them 14 by 12 inches in section, to carry a line of rails, with the load; and the other two, at the sides, 14 by 7 inches in section. They are floored with 3-inch planks, and are trussed with 1¼-inch rods; they are carried on a pair of cast-iron beams, each with two eyes at the ends, bored out to embrace and slide upon the four upright shafts.

The hoist at the Glasgow terminus of the Glasgow and South-Western Railway has a lift of 20 feet from the floor of the warehouse to the level of the rails. It is similar in design and construction to that of the Great Eastern Railway. It is worked by a six horse-power steam-engine, and lifts from 8 to 9 tons in one minute.

II. *Water Supply.*—Water-tank, pumping machinery, and water-crane. Plate LVIII.

The water-tanks erected for the supply of water to locomotives are usually rectangular, from 5 to 9 feet deep, formed of cast-iron plates, from ½-inch to ⅝-inch thick, bolted together. They are supported on stone or bricks, or on pillars, and are placed at some height above the level of the rails—say, from 12 to 15 feet—to afford a head of pressure sufficient to fill the tender quickly. The bottom of the tank is supported on a row of cast-iron beams, one under each line of seams. Tanks are filled either by gravitation or by pumping from a lower level,—the latter most commonly; and in this case the pump and the engine, and boiler, if any, are usually housed in the space under the tank. The tank and machinery, illustrated by Plate LVIII., are examples of the usual appliances. The tank is 21 feet square and 8 feet 6 inches high, constructed with plates 4 feet square and ¾-inch thick, with 6-inch corner-plates. The flanges of the plates are ¾-inch thick, and united by ¾-inch bolts and nuts. The sides are stayed by tie-rods across the tank from side to side. The tank is supported on the walls of the tank-house, at a level of 15 feet above the rails to the lower side of the tank. It is also carried on four cast-iron girders resting on the walls, having a span of 16 feet, and being 16 inches deep at the centre. A 3-inch cast-iron overflow pipe, with a bell-mouth, is fixed inside the tank, the lower end reaching into the well.

The steam-engine and boiler are of the vertical kind, united and placed on one base. The boiler is cylindrical, and is 2 feet 8 inches in diameter and 6 feet 6 inches high. It has an internal circular firebox, with nineteen 2¾-inch wrought-iron flue tubes. The upper hood is of cast-iron, to which is connected the chimney, of sheet-iron, 9 inches in diameter, diverted so as to pass upwards outside the building and alongside the tank. The shell or barrel of the boiler is united to a base of cast-iron by an angle-iron junction. The base is hollow, and is employed as a reservoir to hold the feed-water. The engine has one steam cylinder 7 inches in diameter, with a stroke of 12 inches. The crank-shaft is of hammered iron. The feed-pump is fitted with a suction-pipe of wired India-rubber and galvanized iron strainer.

The pumps consist of two double-acting barrels, 6 inches in diameter, lined with brass, with 18 inches of stroke. The clack-valves are leathered, with copper bucket-rods. There is a separate retaining-valve, with doors for inspection, and a cast-iron air-vessel with galvanized iron dip-pipe. The suction and delivery pipes are 4 inches in diameter, and are fitted with a strainer at the lower end; the example illustrated shows the arrangement for a well 40 feet deep. The water is delivered through the bottom of the tank with a syphon pipe inside. The pumps are worked from the engine by the medium of a spur-wheel and pinion, and with 1¼-inch rods. They are placed at about 17 feet down the well.

The water is drawn from the tank at the bottom through a cast-iron pipe 6 inches in diameter, leading to the water-crane. The pipe is closed by a valve at the mouth, which is lifted for the discharge of water by means of a lever pivotted to the edge of the tank, with a chain hanging down outside within reach of the hand.

The water-crane, shown in connection with the tank, is of the usual construction, consisting of a swing-pipe, balanced on a vertical pivot, within a cast-iron column, so as to swing horizontally into any convenient position to deliver water to the tender. A leather hose is fixed to the extremity of the swing-pipe. A shut-off screw-valve is placed on the pipe behind the crane, by which the water can be cut off when a sufficient supply has been drawn off.

A pillar fountain is shown in the same general view, in advance of the water-crane, with a top and a flexible hose-pipe for the supply of water for general purposes.

On the same plate, a swing water-crane is shown, to a larger scale, in elevation and section. In the section, it is seen that the pivot has a brass bearing in a socket bored out to fit; the socket being connected to the supply-pipe, which ascends from below the level of the ground within the column. The column is heated artificially in winter by a small fire within the base, for which air is admitted through an aperture below. Exit for the products of combustion is provided at the upper end through a number of small apertures. Thus the freezing of the water in the crane is prevented.

The wall water-crane illustrated is employed when the tank-house is conveniently situated for its being applied. It has its bearings at the bottom on a bracket bolted to the wall. At the top it is maintained by the supply-pipe, on which it is pivotted. The water is taken direct from the tank at the bottom, where a sluice-valve is applied, which is worked from the extremity of the swing-pipe.

III.—*Switches and Crossings.*—Wild and Parson's switch, Parson's reversible steel crossing. Plate LIX.

Switches and crossings are usually constructed of the ordinary rails in use on the line. But more recently, the manufacture of special rails for their construction has been adopted. It need hardly be explained that the object of switches and crossings is to afford a means of transferring engines and trains from one line to another, and into sidings. Switches are made either double or single,—that is to say, with two movable points, or only one movable point. The former admits of the better adjustment of the rails in changing the way, and affords a greater extent of bearing surface at the gaps, where a deficiency is more or less unavoidable. They are, therefore, always used on main lines. The single point is used on sidings.

Switches constructed of ordinary double-headed rails are open to the objection of the insufficient wearing surface of the lower table of the rail on the chairs, and to the instability arising from the great height of the tongue-rail relatively to the width of its base. Hence the adoption of specially-formed rails for the construction of the points. Wild and Parson's switch, illustrated in Plate LIX., is on a very good model, and removes the objections to switches constructed from ordinary rails. The tongue-rails have great width of base—nearly 4 inches,—whilst the height is only 3¾ inches, as against 5 inches, the height of the fixed rail. Thus, it has both stability and lateral strength. Again, the tongue is so tapered that the end of it is housed under, instead of being notched into, the upper table of the fixed rail; thus, the train is transferred smoothly and equally from the fixed line of rails to the siding, or *vice versâ*. The fixed rails are secured into the chairs by wedges, instead of by pins. The rods from the hand-lever for moving the switches are passed under the fixed rails, the sliding surfaces of the chairs, on which the switches rest, being necessarily elevated above the other portions of the base of the chairs. They are easily oiled and cleaned, and are less liable to get choked with ballast, and to derange the action of the switches, than when all on one level, as in common switches, made from the ordinary rails. The switch-box contains the weight employed to load the lever, so as, when free, to sit and to hold the switches in the position, open or shut, in which they are required to remain when at rest. The tongue-rails are each 12 feet long. There are two heel-chairs, to which the switches are pinned, and on which they turn, and eight intermediate chairs, on which they slide.

Parson's reversible steel crossing is constructed of steel rails, similar in section to an ordinary double-headed rail, except that the upper table is formed square at one side. This formation adds to the bearing surface of the rail where it is most needed—at the gap—the wheels being thus well supported by the wing-rails before quitting the point, and passing smoothly over the gap. The point-rail, *a*, is planed to the form shown, and the end of the side-rail, *b*, is bound between the tables of the rail *a*, instead of being notched into it, as in ordinary crossings. The wing-rails, *c*, *c*, are firmly maintained in their relative positions, with respect to the point and side-rails, by fishing-blocks of cast-iron, *d*, interposed between them, the whole being united by through-bolts and nuts. The rails rest directly on the sleepers, without the intervention of chairs, and are secured to them by brackets, *e*, with distance-pieces, *f*;—altogether giving a large area of bearing surface on the rails. The rails can be made of any length, and thus a firm connection may be made with the permanent way, and the injurious rocking motion to which short cast-iron crossings are liable is altogether obviated. The greater durability of this crossing, as compared with the common crossing, has been well proved in practice.

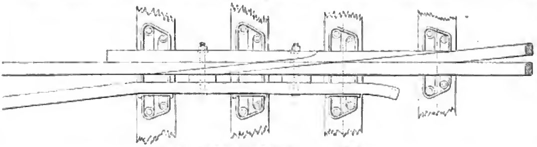


Fig. 233.—Continuous Crossing, by Mr. R. Price Williams.

Mr. R. Price Williams has designed a "continuous crossing," shown in plan, Fig. 233, composed of rails bolted together, in which the gap is formed by planing out of the solid. This crossing comprises three pieces of rail, as against four pieces in Parson's crossings; but it is defective in the formation of the wing, which is planed out of the solid, as it wants the desirable "sweetness" of curvature, which is very properly conferred upon the independent wing-rail.

ADDENDA.

RADIAL LOCOMOTIVE, BY MR. JOHN CLARK.

Mr. Clark, who is favourably known as the inventor of Wilkin and Clark's brake, which has been described in connection with the carriage stock of the North London Railway, has recently (November, 1870) designed a "radial engine," and radial axles for long carriages, for the purpose of facilitating the passage of rolling stock on curves (Fig. 233). It is now clearly understood, and generally admitted, that curves on railways materially augment the resistance to the passage of engines and trains, as compared with the normal resistance on straight lines of rail; which has been fully shown in the body of the text, in the chapter on Resistance of Trains (p. 289). It has been Mr. Clark's object, in designing his radial stock, to reduce such augmentation of resistance.

In designing the engine, Mr. Clark desired to secure the greatest amount of adhesive weight, without placing an excessive load upon any of the axles. Take a tank engine, for example (Fig. 234), having five axles, and weighing 40 tons in working order. The load carried by each axle, supposing the weight to be equally distributed, would be 8 tons, which could be easily carried on rails weighing 50 lbs. per yard. The leading and trailing axles radiate to the curve, and thus the fixed wheel-base is limited to the extreme distance of the three central axles, enabling the engine to pass without difficulty round curves of 3 chains radius. The two side tanks, according to this design, contain 1,200 gallons; and the coal-box holds 3½ tons of coal. The annexed figures (Figs. 235 and 236) show one of the radial axles in plan and elevation, the axle consisting, in fact, of two axles, one within the other. The inner or solid axle runs in bearings on the frame of the engine, like the bearings of the other axles, and is coupled to them as a parallel axle. The outer or hollow axle is pivotted upon the solid axle, at the middle of its length, working as on a universal joint; the wheels are made fast to it, and it is embraced by a triangular bracket, taking its bearings on the axle behind the wheels, and working on a pivot fixed to the frame of the engine, having a movement like that of the Bissell system, as applied to a single pair of wheels. From this it appears that, whilst the outer axle, with the wheels, is free to radiate to the curves, the inner axle works constantly parallel to the other axles, to which it is connected by ordinary coupling-rods. The outer axle is turned by the

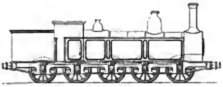


Fig. 234.

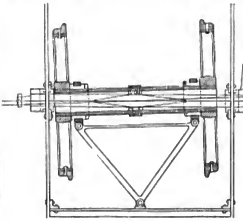


Fig. 235.

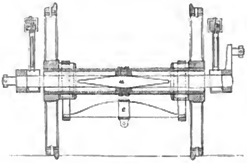


Fig. 236.

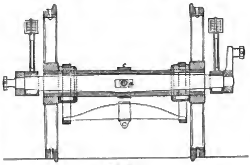


Fig. 237.

inner axle, which, for this purpose, is formed square at the middle, and is embraced by a square thimble or bush, fixed inside the hollow axle, which, though bound to revolve, is free to slide laterally on the fixed axle, for radial adjustment.

An alternative form of connection between the inner and outer axles is shown in Fig. 237: a pin or bolt is passed right through the two axles at the middle, the outer axle being formed with oblong slots, to permit, as before, of lateral movements for radial adjustment.

The two forms of radial axle are shown in section by Figs. 238 and 239.

Fig. 238.

Fig. 239.

The load is proposed to be applied to the hollow axle, as shown in the figures, by means of a transverse spring underhung, the ends of which are linked to the bearings inside the wheels, the weight being applied at the middle.

This class of engine is promising, and would be suited to work on cheaply-constructed lines, having steep gradients, sharp curves, and light rails.

RADIAL AXLES FOR LONG CARRIAGES, BY MR. JOHN CLARK.

This arrangement of self-acting radial axles is applicable to six-wheeled carriages or waggons; and it acts by means of the pressure of the rails on the wheels, and the deflection of the axles, due to the curvature. The middle axle is fitted with quadrants, or bell-crank "wings" or levers, one on each side of the axle-boxes, and linked to them. The bell-cranks are also linked to the extreme axle-boxes on each side respectively, whilst they work on the middle axle-guards as fulcra. When a carriage thus fitted runs on a straight line, the three axles are maintained by their connections truly parallel; on entering a curve, the leading and trailing axles are deflected inwards, and the middle axle outwards, and the outer ends of the extreme axles are thrust apart by means of the bell-cranks, so as to radiate them to the curve. The application of this system to a long carriage, on the 3 feet 6 inch gauge, for Canada, is exemplified in Fig. 240. The carriage is 35 feet 3 inches long, and is constructed to pass freely round curves of a minimum radius of 5 chains. On a 5-chain curve, the middle pair of wheels are carried laterally under the saddle-plates, fitted to slide in the axle-guards (supporting the side-springs), to the extent of 2¼ inches. By means of the bell-cranks the extreme axles are each moved 2 inches out of the square, so as to radiate to the curve. The middle axle-guards are made wider than usual, and rounded on the inner edges. Tubular thrust-rods are employed to connect the bell-cranks with the extreme axle-boxes. It is reported that the carriages run unusually steady. A central buffing and drawing arrangement is applied at each end of the carriage; and the coupling, as well as the detail of the side-springs, are similar to those of the Norwegian carriages, already described and illustrated, p.309.

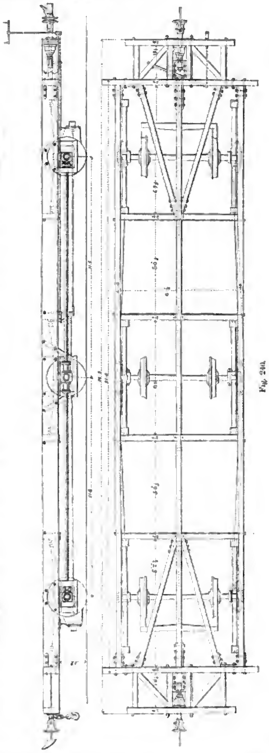


Fig. 240.

The radial axles have been applied to stock on the Mont Cenis, the Queensland Great Northern, the Queensland Southern and Western, the Honduras, and the Coquimbo (Chili) Railways; and the official accounts of the performance of the stock so fitted are very favourable. From Coquimbo the Manager reports, that "The new radiating axles in the carriages and waggons answer admirably. They are the smoothest running stock in Chili beyond comparison." "The carriages which have been altered [to be fitted with the radial axles] are those which formerly had the two-wheeled bogies. They were so unsteady, and jolted so much, that I was obliged to take them off the road. They are now perfectly easy; and at thirty-five miles per hour (a great speed for us), there was no perceptible oscillation. The same remark applies to the new waggons. On our return journey, we brought with us two of the old and one of the new waggons. The oscillation in the former was so great, that, had they not been loaded, I think they would have run off the line, while with the latter there was no motion apparent."

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